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IMPROVING ENGINE PERFORMANCE BY EXHAUST PIPE TUNING.

BY P. H. SCHWEITZER, LT. COMMANDER, U.S.N.R.

Perhaps the most neglected phase of two-stroke cycle engine design pertains to the exhaust systems. This neglect is fully reflected in the almost complete absence of American literature on the subject. This is the more deplorable as a properly designed exhaust system does not usually involve added weight or added complications and yet it improves engine performance all around. Even engines already installed could be improved by "tuning" their exhaust. Exhaust system design with regard to pressure waves is now in a status analogous to that of crankshaft design 20 years ago with regard to torsional vibrations. Considerable information has been discovered, but it has not

spread widely enough and is generally ignored by the practical designers and installation men.

To be on the "safe" side, exhaust pipes are frequently oversized. But an oversize exhaust pipe is no better guarantee against undesirable synchronism than an oversize crankshaft is. Synchronism in torsional vibrations results in crankshaft breakage. Synchronism in pressure waves results in poor scavenging, reduced air charge, low power output, high fuel consumption and high exhaust temperatures. By avoiding the "criticals" the crankshaft is safe from torsional failure. By avoiding synchronism of pressure waves with engine speed we avoid poor filling of the cylinder and resultant power loss. But we can go one better by harnessing the pressure waves of appropriate frequency and thereby gain powerful assistance in scavenging and charging the cylinder. The result is higher output, lower fuel consumption, lower exhaust temperatures, lower piston and cylinder head temperatures, reduced maintenance and longer life. The difference between a well tuned and poorly tuned exhaust system is sometimes 30 per cent in engine output.

The exhaust system is frequently evaluated on the basis of the exhaust back pressure. But the exhaust back pressure as read on a manometer or pressure gauge attached to a certain point of the exhaust pipe is not significant because the actual pressure at any one point fluctuates during the cycle and the reading is only a rough average. Yet it is not this average which controls the charging and scavenging but the actual pressures during the scavenging period. The engine which has low (still better, sub-atmospheric) exhaust pressures during most of the scavenging period will give a better performance than one with high exhaust pressures, although the manometer reading of the exhaust pressure may be the same.

If no exhaust pipe were used, evidently the pressure at the exhaust opening would constantly be atmospheric. If a pipe is used, the sharp impulse initiated when the high pressure cylinder gases are put in communication with the exhaust pipe, sets up pressure waves of appreciable magnitude which travel back and forth in the pipe with the velocity of sound. The pressure next to the exhaust opening will rise and fall in accordance with the natural frequency of the system but with declining ampli-

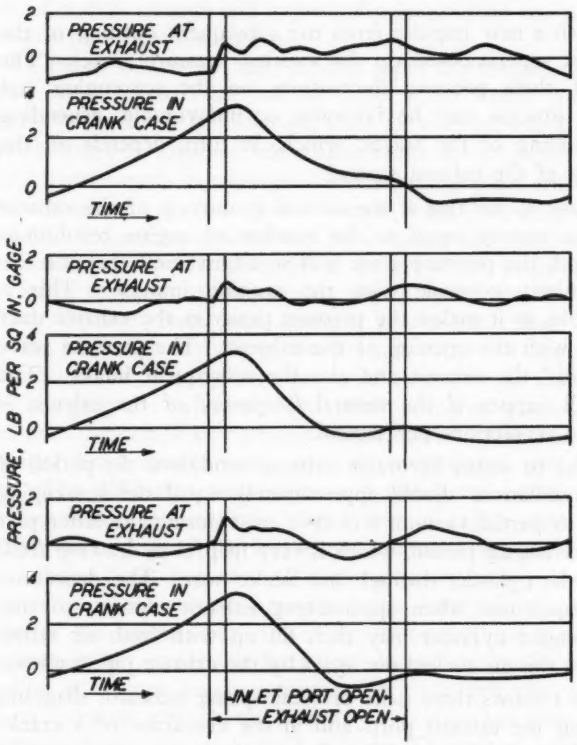
tudes until a new impulse from the subsequent opening of the exhaust is superimposed on the existing pressure waves. The effect of these pressure fluctuations on the scavenging and charging process may be favorable or unfavorable, depending on the timing of the waves, which, in turn, depends on the geometry of the exhaust system.

It is easy to see that if the natural frequency of the exhaust column is exactly equal to the number of engine revolutions per second, the pressure wave will so adjust itself that the rise will regularly coincide with the exciting impulse. This is undesirable, as it makes the pressure peaks in the exhaust duct coincide with the opening of the exhaust. The pressure wave will "buck" the exhaust and also the subsequent intake. The same will happen if the natural frequency of the exhaust is twice the revolutions per second.

In order to secure favorable exhaust conditions the period of natural oscillations should approximately equal the scavenging period. A partial vacuum will then exist during the latter part of the scavenging period, which is very helpful in drawing fresh air into the cylinder through the intake port. The depression should taper out when approaching exhaust closure, so that the scavenged cylinder may then fill up with fresh air rather than have this air sucked out again by the exhaust pipe vacuum.

Figure 1 shows three pairs of weak-spring indicator diagrams taken near the exhaust ports and in the crankcase of a crankcase-scavenged engine. The first pair shows unsatisfactory scavenging. Since the pressure never drops below 0.4 pounds per square inch gauge during the whole scavenging period, the scavenged air cannot expand lower than to this value. Only after the closing of the ports does the pressure in the crankcase drop sufficiently to draw in fresh air from the outside. The pressure drop between crankcase and exhaust port is small or negative during the second half of the charging period which resulted in poor volumetric efficiency.

The third (bottom) pair of diagrams show very good scavenging. Note the vacuum at the exhaust near port closure. This causes a corresponding vacuum in the crankcase. The effect is that fresh air from the outside is drawn vigorously



(Schmidt)

FIGURE 1.—EFFECT OF TUNING ON PRESSURE FLUCTUATIONS IN THE EXHAUST PIPE AT THE EXHAUST PORT AND IN THE CRANKCASE OF A CRANKCASE SCAVENGED ENGINE DURING ONE CYCLE. TOP DIAGRAMS SHOW POOR, BOTTOM DIAGRAMS GOOD, MIDDLE DIAGRAMS INDIFFERENT TUNING.

through the engine. The absolute exhaust pressure begins to rise when the piston is approaching port closure and depression. That helps to fill up the cylinder by the ramming effect of the air. The resulting high volumetric efficiency raised the thermal efficiency from 0.222 in the top diagrams to 0.252, an increase of 13.5 per cent. In the middle pair of diagrams conditions are in between. The only difference in the engine condi-

tions represented by the diagrams was in the tuning of the exhaust system.

It may be in order to point out here that the "tuning" of the exhaust system depends primarily on its geometry, that means on the length and diameter of the exhaust pipes and on the volume of the various containers attached or interposed in the exhaust system. If an exhaust system is tuned it stays tuned irrespective of load and operating conditions. The only exception is a change in speed. An engine can be tuned for one speed only, and, therefore, tuning has the greatest significance for constant speed engines. Variable speed engines should be tuned to the speed at which optimum performance is desired. This may be the most common operating speed or the speed of maximum power output.

Being aware of the importance of the exhaust tuning, the practical problem is how to create favorable exhaust conditions for a given engine at a given speed. Three methods will be described in this article: 1. Performance test method; 2. Analytical method; 3. Pressure indicating method. Often a combination of the methods will be found advantageous.

Performance Test Method.

Exhaust tuning affects such important performance characteristics as maximum power output, specific fuel economy, air delivery and exhaust temperature. In a most direct method, therefore, we would measure some such performance characteristics while the tuning of the exhaust system is being varied. Then we select the tuning which gives best engine performance.

This method can be applied very successfully to a small engine. Belilove (Ref. 1) measured the air delivery ratio (volume of air delivered to the cylinder divided by the piston displacement) and the power output of a 3/4 Hp. Evinrude outboard engine while he varied the length of the 3/4 inch exhaust pipe (inside diameter 13/16 inch), and obtained Figure 2 (6). It was simple to conclude that a 28 inch long pipe was the best for that engine at 2800 Rpm.

For large engines it is not always feasible to measure power, air consumption or specific fuel consumption because of lack

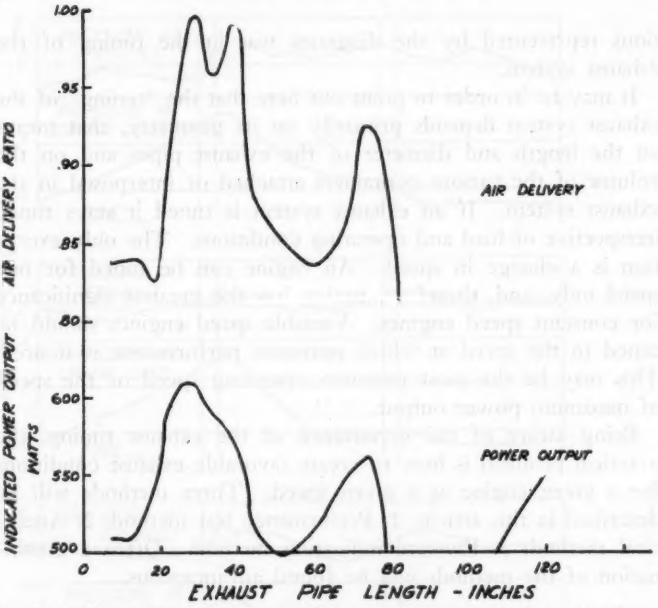


FIGURE 2.—POWER AND AIR DELIVERY WITH AN EVINRUDE 1⅛-INCH×1⅛-INCH SPARK IGNITION ENGINE AT 2800 RPM.

of equipment. The measuring of the exhaust temperature alone without power measurement is inadequate. Making numerous changes on a bulky exhaust system is costly and time-consuming. Another unattractive feature of this method is that one or two measurements do not disclose how near the system is to optimum tuning and in which direction the optimum will be found. Therefore, the principal role of this method lies in checking the results obtained by other means.

ANALYTICAL METHOD.

Taking advantage of such theoretical work as was done by Schmidt (Ref. 2) and Zeman (Ref. 3) in Germany, and Farmer (Ref. 4) and Mucklow (Ref. 5) in England, the problem of exhaust tuning can be made accessible to analytical calculations. In this manner the exhaust system can be tuned before it is

built, saving much time and expense. In the following, methods of calculating exhaust systems are to be presented in a simplified manner, ignoring the derivations of the formulae and preceding theoretical work.

The frequency of the gas column oscillations in the exhaust system is determined primarily by the length of the exhaust pipe and secondarily by its diameter and the volumes interposed in the system. Therefore, it is reasonable to start the calculations by selecting the diameter of the main exhaust pipe. If the exhaust pipe cross section is inadequate the exhaust will be throttled irrespective of its tuning. If it is too large, the amplitude of the pressure waves will be small and the effect of tuning will thereby be minimized. That is an advantage if the tuning is incorrect but a disadvantage if it is correct. The tendency in the past has been to use oversized exhaust pipes for two-stroke cycle engines, presumably just to be on the safe side and avoid the effects of the "incalculable" pressure waves.

There is little beyond thumb rules to guide us in the selection of the diameter of the exhaust pipe. To avoid throttling, the gas velocity in the pipe or duct must be lower than in the exhaust ports, preferably a third less. In multi-cylinder engines the gas velocity in the common header or exhaust pipe should be still lower. Burgess Battery Co., makers of exhaust snubbers, recommend 50 feet per second for crankcase scavenged engines; from 65 to 115 feet per second for low-speed (up to 350 Rpm.) separately scavenged engines; from 100 to 150 feet per second for medium-speed (350 to 1200 Rpm.); and from 135 to 165 feet per second for high-speed (above 1200 Rpm.) two-stroke cycle engines. In calculating the conduit size from the permissible gas velocities, it must be taken into account that the volume of the exhaust gas is about double the volume of the intake air, based on the ratio of absolute temperatures.

Example: A 16-cylinder 8-1/2x10-inch 800 Rpm. engine has an air-delivery ratio of 1.325 and a gas temperature (in the exhaust pipe) of 500 degrees F. at full load. What is the required size for the exhaust pipe? The exhaust gas volume is

$$1.325\times16\times\frac{8.5^2\pi}{4}\times10\times\frac{800}{1728}\times\frac{460+500}{460+150}=8700$$

cubic feet per minute.

Allowing an average gas velocity of 150 feet per second, the cross section of the common exhaust pipe must be

$$A=\frac{8700\times1728}{60\times150\times12}=139 \text{ square inches}$$

so that the inside diameter of the conduit is to be 13.3 inches. The next standard pipe size is 14 inches nominal inside diameter.

After the size of the exhaust pipe has been selected, its tuning will be effected by adjusting its length and the volumes interposed in the system. The controlling factor in the calculations is the natural frequency of the pressure waves.

Pressure Waves.

The pressure waves in the exhaust pipes are similar to the sound waves in organ pipes and are controlled by identical laws. In a plain pipe closed at both ends the period of the pressure waves is 2 L/a where L is the length of the pipe and a is the velocity of the sound in the gas. The value of a varies with the gas temperature according to the formula

$$a=C\sqrt{k\,p\,v}=C\sqrt{k\,R\,T}$$

where k is the ratio of the specific heats of the gas, p, v, and T its mean pressure, specific volume and absolute temperature in the pipe, and C a constant which decreases in straight line relationship inversely as the bore of the pipe. According to the most recent determinations (Ref. 1) Figure 3 represents the velocity of pressure propagation in pipes with average composition of exhaust gas for various mean exhaust gas temperatures and pipe diameters. The statement on page 189 that the tuning is independent of the load needs correction in so far that a change in load changes the exhaust temperature which changes to some extent the propagation velocity of the pressure wave.

For an exhaust system consisting of a single pipe of uniform cross section attached at one end of the exhaust ports, the other end being open to the atmosphere, the period of gas column vibration is about double, that means 4 L/a. The reason for this is that the pressure wave is reflected at the open end of the pipe with sign reversed. The period of this negative wave is

also 2 L/a and, therefore, the total time of the complete cycle is 4 L/a.

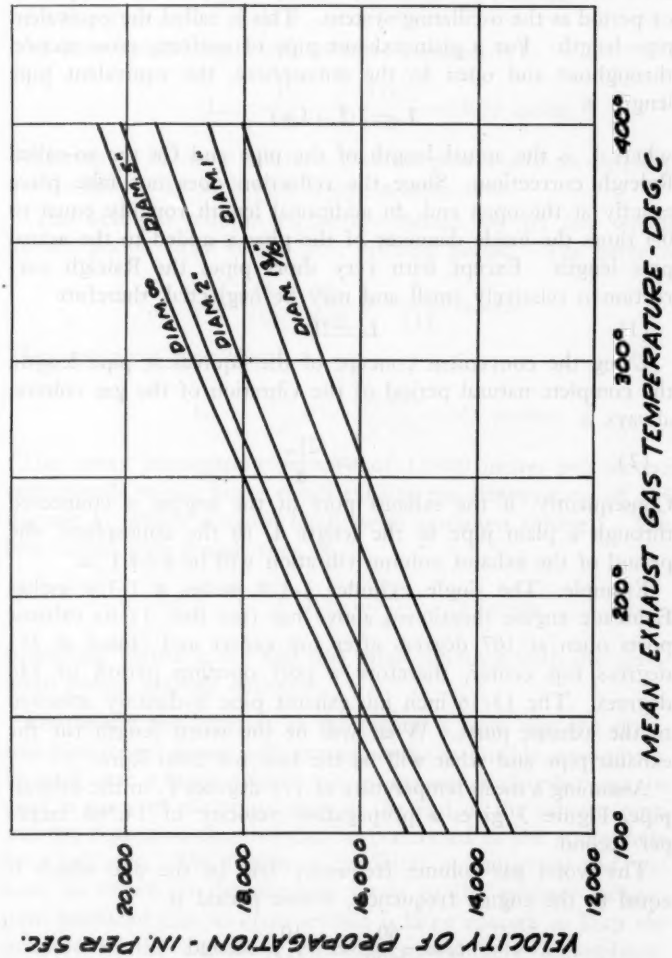


FIGURE 3.—VARIATION OF PROPAGATION VELOCITY OF PRESSURE WAVES IN PIPES OF VARIOUS DIAMETERS.

In calculation of pressure waves in pipes it is customary to replace the oscillating system with a pipe of length L_e of uniform cross section closed at both ends which has the same frequency or period as the oscillating system. This is called the equivalent pipe length. For a plain exhaust pipe of uniform cross section throughout and open to the atmosphere, the equivalent pipe length is

$$L_e = 2(L + C_R)$$

where L is the actual length of the pipe and C_R the so-called Raleigh correction. Since the reflection does not take place exactly at the open end, an additional length roughly equal to 0.4 times the inside diameter of the pipe is added to the actual pipe length. Except with very short pipes the Raleigh correction is relatively small and may be neglected, therefore

(1) $$L_e = 2L$$

Using the convenient concept of the equivalent pipe length, the complete natural period of the vibration of the gas column always is

(2) $$t = \frac{2L_e}{a}$$

Consequently, if the exhaust port of the engine is connected through a plain pipe of the length L to the atmosphere, the period of the exhaust column vibration will be $t = 4\ L/a$.

Example: The single cylinder 1-3/8 inches x 1-3/8 inches Evinrude engine mentioned above has (see Ref. 1) its exhaust ports open at 107 degrees after top center and closed at 253 degrees top center, therefore a port opening period of 146 degrees. The 13/16 inch i.d. exhaust pipe is directly attached to the exhaust ports. What will be the worst length for the exhaust pipe and what will be the best for 2800 Rpm?

Assuming a mean temperature of 175 degrees F. in the exhaust pipe, Figure 3 gives a propagation velocity of 14,700 inches per second.

The worst gas column frequency will be the one which is equal to the engine frequency, whose period is

$$t_w = \frac{60}{2800} = \frac{10}{465} \text{ second}$$

The equivalent pipe length which gives this frequency to the

system is from Equation (2)

$$L_{ew}=\frac{a\,t_w}{2}=\frac{147000}{2\times 465}=158.5 \text{ inches}$$

The actual pipe length will be from Equation (1)

$$L_w=\frac{L_{ew}}{2}=\frac{158.5}{2}=79.25 \text{ inches}$$

Referring to Figure 2 we note that the power and air delivery curves take at 80 inch pipe length a steep dive.

The best gas column frequency should be the one which gives a period equal to the port opening period. The latter is

$$t_b=\frac{146}{360}\ \frac{60}{2800}=\frac{1}{111}\text{second}$$

The equivalent pipe length is

$$L_{eb}=\frac{a\,t_b}{2}=\frac{13800}{2\times 111}=62 \text{ inches}$$

(The lower propagation velocity of 13,800 inches per second corresponds to the lower mean exhaust gas temperature of 125 degrees F. which is more likely with optimum tuning.) The best actual pipe length then is

$$L_b=\frac{62}{2}=31 \text{ inches}$$

Inspecting Figure 2, we find this value to check with the maximum power and air delivery.

Frequently the location of the engine is such that the desirable pipe length is not sufficient to reach the outside atmosphere. In such case a large exhaust pit or expansion chamber can be used in place of the atmosphere at the end of the exhaust pipe, and the expansion chamber can be connected to the atmosphere by a tail pipe. The length of the latter is immaterial and will have no effect on the pressure waves in the primary exhaust pipe, provided that its cross section is large enough to keep the pressure in the expansion chamber substantially atmospheric. A rule will be given later in the text for the size of the expansion chamber and the minimum diameter of the tail pipe.

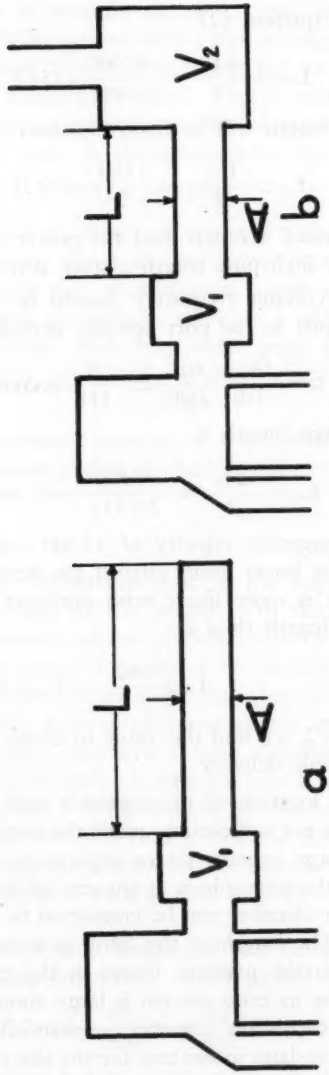


FIGURE 4.—ARRANGEMENT OF EXHAUST SYSTEMS.

However, ordinarily the exhaust pipe does not connect to the exhaust ports directly, but a duct, sleeve or chamber of a certain volume is placed between the exhaust ports and the exhaust pipe. The arrangement is shown in Figure 4 (a) when the exhaust connects directly into the open and Figure 4 (b) when it connects into an expansion chamber. The volume V_1 next to the exhaust ports will affect the exhaust tuning in either case and we will refer to it as the "exhaust pot" even if it consists only of a small enlargement over the exhaust pipe cross section. The size of the expansion chamber V_2 and tail pipe does not affect the tuning if both are large enough to keep the pressure in the expansion chamber substantially atmospheric.

The equivalent pipe length of such a system has been calculated by Thomas Schmidt (Ref. 2) and can be expressed by the following formula:

$$\tan\frac{\pi L}{L_e}=\frac{A\,L_e}{\pi V_1} \qquad (3)$$

where L is the actual, L_e the equivalent pipe length, A is the cross sectional area of the exhaust pipe and V_1 the volume of the exhaust pot close to the engine. Chart, Figure 5, shows the relation in graphical form.

It will be noted that equivalent pipe length (and therefore, the natural frequency of the exhaust system) is determined not by the pipe length alone, but also by its cross section and the volume between the engine and the exhaust pipe. Even a small volume V_1 increases the equivalent pipe length considerably. For instance, in the example treated above an exhaust pot of only 26 cubic inches between the engine and a 25 inch exhaust pipe will increase the equivalent pipe length from 51 inches to 121 inches.

In using Formula (3) or Chart 5, the exhaust pot volume V_1 should include any enlargement found beyond the exhaust port such as ducts, sleeves, etc., but only the volume in excess of the corresponding exhaust pipe should be counted. The actual exhaust pipe length should be counted from the cylinder to the atmosphere or the large expansion chamber, and to that the Raleigh correction of 0.4×I.D. may be added.

Example: A 7.9×11.8-inch, one cylinder, 18-Hp. crankcase-

scavenged engine operating normally at 370 Rpm. has an exhaust opening period of 136 degrees crank angle. The exhaust ports connect directly into an exhaust pot of 2000 cubic inch volume. From this an exhaust pipe of 5.35 inches I.D. leads to the atmosphere. What is the worst exhaust-pipe length, and what is the best?

The worst frequency is that equal to the engine frequency which corresponds to a period of

$$t_w = \frac{60}{370} = \frac{1}{6.15} \text{second}$$

The corresponding equivalent pipe length is from Formula (2)

$$L_e = \frac{a\, t_w}{2}$$

With an estimated mean exhaust gas temperature of 210 F., the propagation velocity of the pressure waves from Figure 3 is 14,900 inches per second and, therefore,

$$L_e = \frac{14900}{2 \times 6.15} = 1210 \text{ inches}$$

The ratio $V_1/A = 2000/22.5 = 89$ inches, which gives through Chart 5 an actual pipe length of 515 inches less $0.4 \times 5.35 = 2.14$ inches Raleigh correction, therefore

$$L_w = 513 \text{ inches}$$

This would be the worst pipe length.

The best frequency is that which corresponds to the period of exhaust duration, which is

$$t_b = \frac{136}{6 \times 370} = \frac{1}{16.33} \text{seconds}$$

and the corresponding equivalent pipe length

$$L_e = \frac{a\, t_b}{2} = \frac{14900}{2 \times 16.33} = 456 \text{ inches}$$

Chart 5 gives now with the same $V_1/A = 89$ inches, 150 inches and applying again the Raleigh correction

$$L_b = 148 \text{ inches}$$

as the best pipe length.

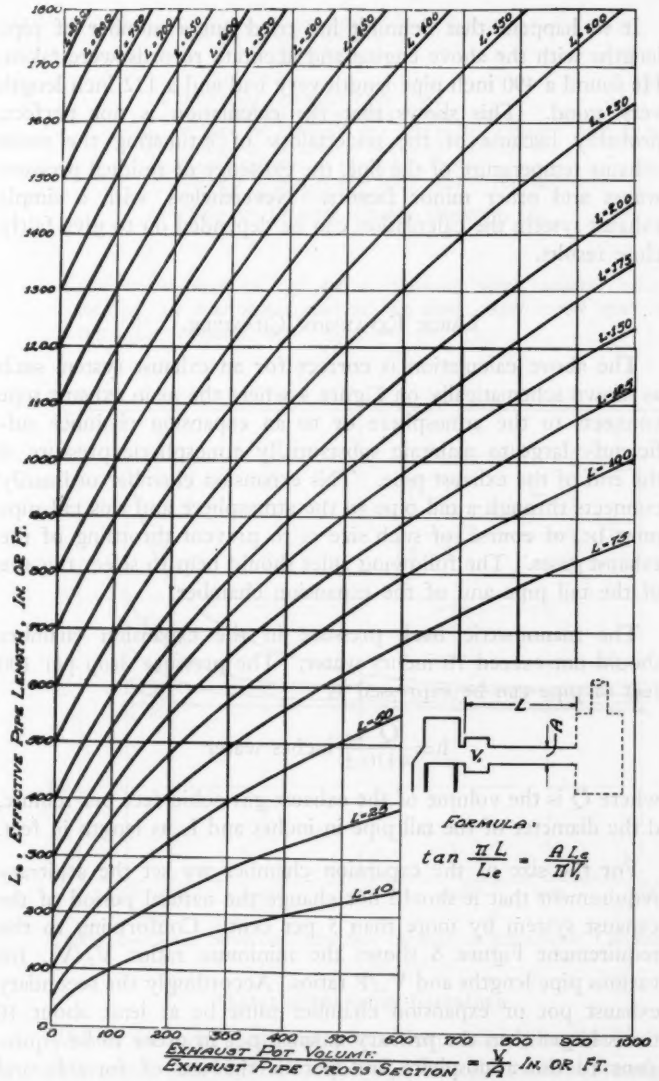


FIGURE 5.—CHARTS SHOWING THE EQUIVALENT PIPE LENGTHS FOR A FREQUENTLY USED EXHAUST SYSTEM.

It so happens that Schmidt has tried out a number of pipe lengths with the above engine and accurate records were taken. He found a 490 inch pipe length very bad and a 132 inch length very good. This shows that the calculation is not perfect, probably because of the uncertainty of estimating the mean exhaust temperature of the line, the existence of residual pressure waves and other minor factors. Nevertheless, with a simple exhaust system the calculation can be depended on to give fairly close results.

Large Expansion Chamber.

The above calculation is correct for an exhaust system such as shown schematically on Figure 4 where the main exhaust pipe connects to the atmosphere or to an expansion chamber sufficiently large to maintain substantially atmospheric pressure at the end of the exhaust pipe. This expansion chamber ordinarily connects through a tail pipe to the atmosphere and this tail pipe must be, of course, of such size as to prevent throttling of the exhaust gases. The following rules should help to select the size of the tail pipe and of the expansion chamber:

The manometric back pressure in the expansion chamber should not exceed 10 inches water. The pressure drop per 100 feet of pipe can be expressed as

$$h=\frac{Q^2 L}{4430\, d^5} \text{ inches water}$$

where Q is the volume of the exhaust gas cubic feet per minute, d the diameter of the tail pipe in inches and L its length in feet.

For the size of the expansion chamber we set the arbitrary requirement that it should not change the natural period of the exhaust system by more than 5 per cent. Conforming to this requirement Figure 6 shows the minimum ratios V_2/V_1, for various pipe lengths and V_1/F ratios. Accordingly the secondary exhaust pot or expansion chamber must be at least about 10 times larger than the primary exhaust pot in order to be equivalent to the atmosphere and permit the use of formula and chart shown in Figure 5.

$$\tan \frac{2\pi L}{a Z_c} = \frac{F/V_p \cdot F/V_s}{\frac{2\pi}{a Z_c} \cdot \frac{a Z_c}{2\pi} \frac{F^2}{V_p V_s}}$$

L = LENGTH OF EXHAUST PIPE, IN.
F = CROSS SECTION OF EXHAUST PIPE, SQ IN.
V_p = VOLUME OF EXHAUST POT NEXT TO EXHAUST PORT, CU IN.
V_s = VOLUME OF SECONDARY EXHAUST POT, CU IN.
a = SOUND VELOCITY IN EXHAUST COLUMN
ASSUMED 17 800 IN. PER SEC = 1480 FT PER SEC
(THIS IS TRUE FOR 484°F)
Z_c = PERIOD OF EXHAUST COLUMN VIBRATION, SEC

FOR GIVEN VALUE OF V_p/F AND L, ORDINATE SHOWS MINIMUM VALUE OF V_s/V_p SO THAT PERIOD DIFFERS NO MORE THAN 5 PER CENT FROM THE VALUE OBTAINED WITH $V_s/F = \infty$

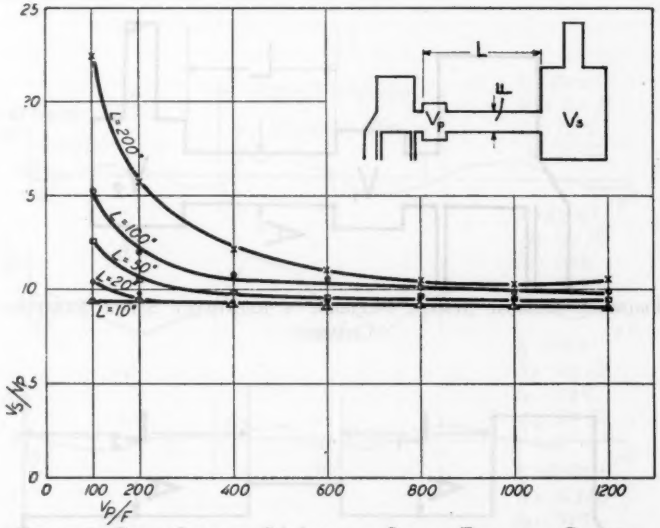


FIGURE 6.—CHART SHOWING SELECTION OF SIZE OF EXPANSION CHAMBER.

SMALL EXPANSION CHAMBER.

When secondary exhaust pot or expansion chamber is relatively small (see Figure 7), it will affect the natural frequency of the oscillating system. Schmidt (Ref. 2) developed the cal-

culation for this case also and the resulting formula is as follows:

$$\tan\frac{\pi L}{L_e} = \frac{\frac{A}{V_1}+\frac{A}{V_2}}{\frac{\pi}{L_e}-\frac{L_e}{\pi}\frac{A^2}{V_1 V_2}}$$

This formula is rather complicated and its use is preferably avoided by increasing the relative size of V_2.

Another system is shown in Figure 8 with a single exhaust pot in the middle of the exhaust pipe, for which the following equation is valid:

$$\operatorname{ctn}\frac{\pi L_2}{L_e}-\tan\frac{\pi L_1}{L_e}=\frac{\pi V}{A L_e}$$

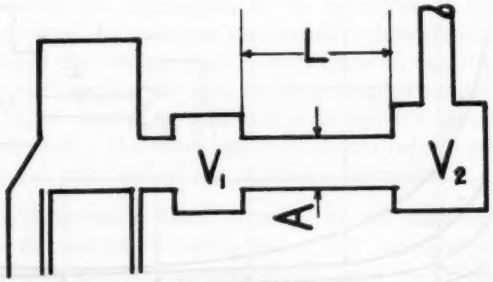


FIGURE 7.—EXHAUST SYSTEM INCLUDING A RELATIVELY SMALL EXPANSION CHAMBER.

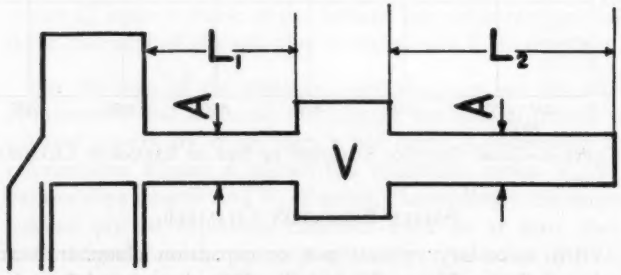


FIGURE 8.—EXHAUST SYSTEM WITH AN EXHAUST POT IN THE MIDDLE OF THE EXHAUST PIPE.

Example: In Figure 9, four pressure diagrams of the exhaust pipe are shown, presenting successively deteriorating tuning. The only variable in the setup was the size of the expansion chamber V_2. In order to equal the performance with no expansion chamber, Schmidt had to supply an expansion chamber of 0.6 m³=36,000 cubic inches. According to Chart 6 the expansion chamber would have to be 17 times the primary exhaust pot or 17×2000 cubic inches=34,000 cubic inches, which checks closely with the experimental value. By reducing the secondary volume to 13,300, 8600 and 1840 cubic inches respectively, diagrams b, c and d were obtained and performance deteriorated, as it is shown by the indicated figures on specific fuel consumption and exhaust temperature. The power output was kept substantially constant during the series.

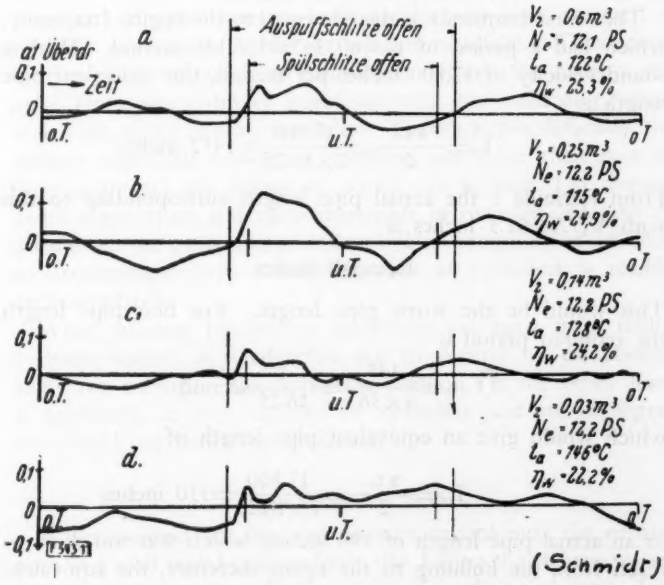


FIGURE 9.—INDICATOR DIAGRAMS SHOWING THE EFFECT OF SIZE OF EXPANSION CHAMBER.

Doubling the Period.

If the optimum exhaust pipe length is too short for structural reasons, even when used in conjunction with an expansion chamber, favorable results can also be obtained by multiplying the equivalent pipe length by two.

Example: A 12×15 inch, 367 Rpm. crankcase scavenged engine had an 8 inch I.D. exhaust pipe discharging into the open. It had a 78-1/2 inch long muffler, close to the engine, with a volume of 8550 cubic inches. The average exhaust gas temperature was estimated at 464 degrees F. and the exhaust per opening period 135 degrees crank angle.

The exhaust-pipe cross section was $8^2\pi/4=50.3$ square inches. Since the muffler was in the line, the replaced pipe volume $50.3\times78.5=3950$ cubic inches had to be deducted, giving an effective exhaust pot volume of $V_1=8550-3950=4600$ cubic inches and $V_1/A=4600/50.3=91.5$ inches.

The worst frequency is the one equal to the engine frequency, which had a period of $t_w=60/367=1/6.125$ second. With a sound velocity of 17,800 inches per second, the equivalent pipe length is

$$L_e=\frac{a\,t_w}{2}=\frac{17{,}800}{2\times6.125}=1452 \text{ inches}$$

From Formula 3 the actual pipe length corresponding to this with $V_1/A=91.5$ inches is

$$L_w=640 \text{ inches}$$

This would be the worst pipe length. For best pipe length the required period is

$$t_b=\frac{135}{6\times367}=\frac{1}{16.25}\text{second}$$

which would give an equivalent pipe length of

$$L_e=\frac{a\,t_b}{2}=\frac{17{,}800}{2\times16.25}=550 \text{ inches}$$

or an actual pipe length of 190 inches, which was too short to reach from the building to the open; therefore, the equivalent pipe length is doubled, making 1100 inches. With this the actual pipe length turns out to be 460 inches as most favorable.

According to tests made by K. C. Whitefield (Ref. 6) on this engine, the worst pipe length was 600 inches and the best 456 inches, which is very good agreement, indeed.

Multicylinder Engines.

The above formulae are strictly applicable only for either single cylinder engines or multicylinder engines with individual exhaust pipes to the atmosphere or to a large expansion chamber. The last arrangement is quite common with an exhaust pit under the floor into which the individual exhaust pipes connect. If the volume of the exhaust pit exceeds the size determined by Figure 6, and it connects with a large enough tail pipe to the open, the multicylinder engine can be treated as so many single cylinder engines as far as exhaust tuning is concerned.

Frequently, however, the individual exhaust pipes combine by tee's or Y's before reaching the atmosphere or expansion chamber. In other cases no individual exhaust pipes are used at all, only a common header or exhaust manifold. These cases are not easily accessible to calculation because the exhaust impulse from one cylinder interferes with the residual pressure waves set up by another cylinder. To evade this difficulty the exhaust manifolds sometimes combine not more than two or three cylinders the firings of which are at least 120 degrees crank angle apart and these manifolds in turn are led into an exhaust pit or expansion chamber. The complication of such an arrangement with a large number of cylinders is readily understandable.

When exhaust headers or manifolds are used in a multicylinder engine, it is advisable not to depend on calculations alone for tuning. The indicating method on the other hand is applicable to any number of cylinders and its intelligent use should pay handsome dividends.

Indicating Method.

It was pointed out that the pressure record near the exhaust port of a well tuned exhaust looks like Figure 9 *a*, where we see a nice negative loop ending at about the exhaust closure. Less good tuning is shown in diagram *b*, where the negative loop is

still pronounced but ends a little too early. In diagram *c* the negative loop almost vanished during the scavenging period, but the worst tuning is shown by diagram *d*, where positive pressure persists during the whole scavenging period.

Therefore, if reliable pressure diagrams of the exhaust of each cylinder are available, by their inspection the quality of the exhaust tuning can easily be determined. If the tuning is not satisfactory, slight changes in the geometry of the exhaust system (size and lengths of ducts and pipes, volumes of ducts, chambers and silencers) should improve it. The shape of the pressure diagram indicates even the direction in which the improvement is to be sought. If the negative loop is too short, the period of oscillation must be increased by adding length and/or volume. If the negative loop is too long, the period must be shortened by decreasing length and/or volume. If for each cylinder the negative loop ends at about exhaust closure, we have perfect tuning.

Taking of exhaust pressure indicator diagrams is not easy with conventional indicators. Mechanical weak-spring indicators are suitable only for relatively low speeds. The diagrams shown in Figure 10 were taken by Belilove (Ref. 1) with a cathode ray oscillograph. An electromagnetic pickup specially fitted with a thin steel diaphragm of 0.0035 inch thickness was attached to the exhaust pipe near the exhaust ports and connected to a 9 inch RCA cathode ray oscillograph to which an integrator and a degree marker was added. The wiring circuit of the integrator and degree marker is shown in Ref. 7.

The records obtained with the Evinrude engine mentioned above check satisfactorily with the performance tests.

Nevertheless it was found that a cathode ray indicator is not very suitable for low pressure indication, especially because of the uncertainty of the pressure scale and the zero line.

On the other hand the instrument shown in Figure 11 has proven itself very satisfactory for indicating exhaust pressures. It has been developed recently in the Diesel Laboratory of The Pennsylvania State College.

Figure 12 is a schematic drawing of the indicator. It consists of a rotor R, with a single distributor hole H, driven from the

engine crankshaft through spline driveshaft D. The rotor rotates inside of stator S which has 36 equally spaced passages P, and brings every ten degrees another manometer tube, M, in connection with the exhaust port. The connection to the exhaust (or other space the pressure of which is to be indicated) is through a tubing attached to the stationary cover plate C, above

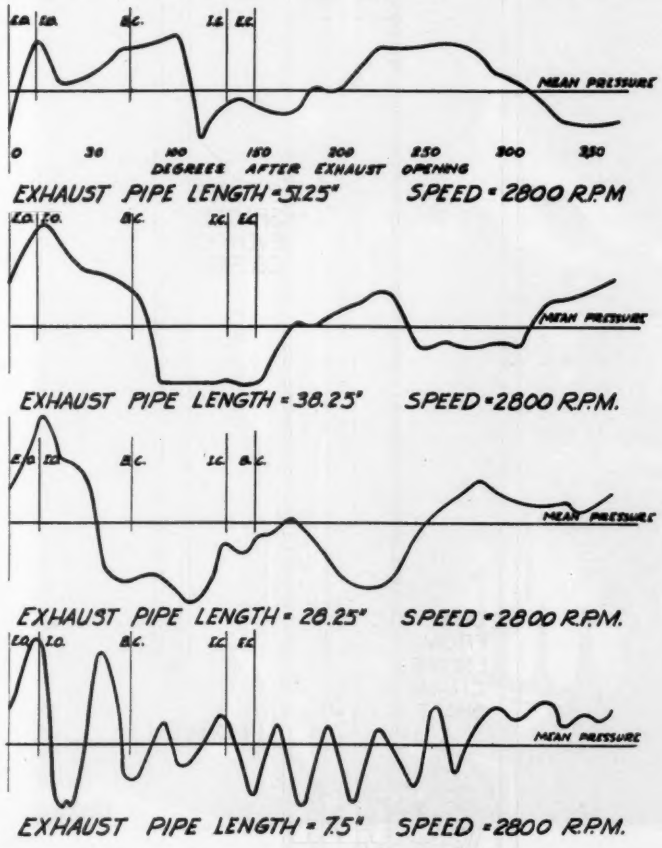

MEAN PRESSURE
0 30 100 150 200 250 300 350
DEGREES AFTER EXHAUST OPENING
EXHAUST PIPE LENGTH = 51.25" SPEED = 2800 R.P.M
MEAN PRESSURE
EXHAUST PIPE LENGTH = 38.25" SPEED = 2800 R.P.M.
MEAN PRESSURE
EXHAUST PIPE LENGTH = 28.25" SPEED = 2800 R.P.M.
MEAN PRESSURE
EXHAUST PIPE LENGTH = 7.5" SPEED = 2800 R.P.M.

FIGURE 10.—PRESSURE TIME DIAGRAMS OBTAINED BY A CATHODE RAY INDICATOR. (THE CARBON PILE PRESSURE ELEMENT WAS PLACED IN THE EXHAUST PIPE NEXT TO THE PORT.)

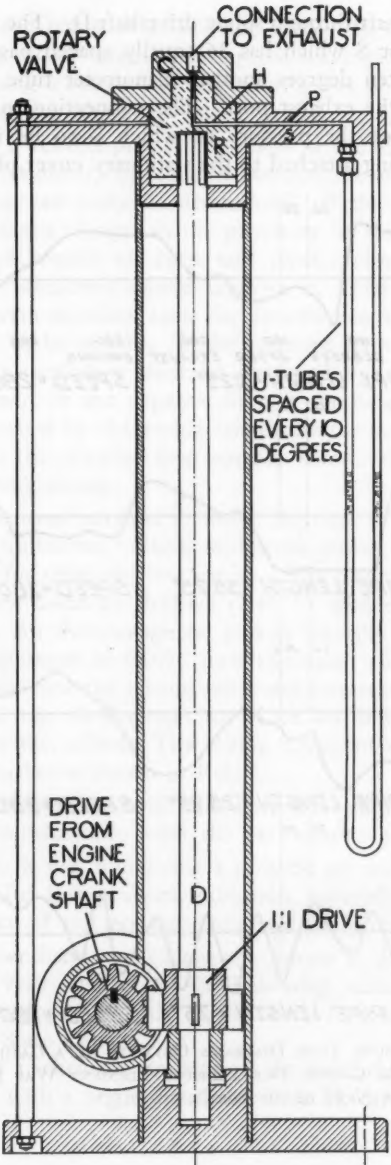


FIGURE 12.—SCHEMATIC DRAWING OF PENN STATE ROTARY VALVE LOW PRESSURE INDICATOR.

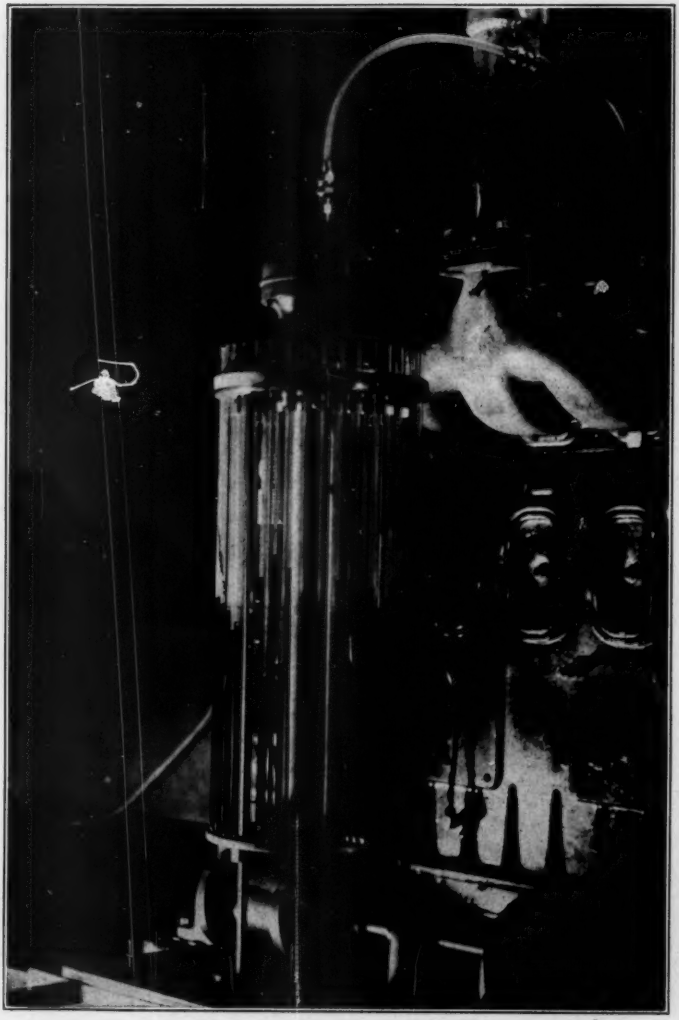
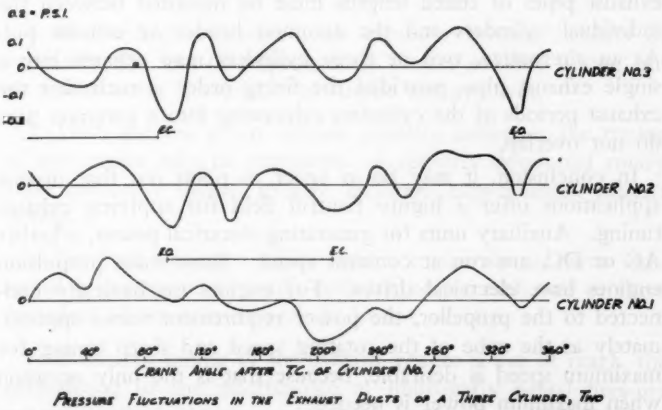


FIGURE 11.—PENN STATE ROTARY VALVE LOW PRESSURE INDICATOR.

rotor R, the distributor hole of which remains in continuous connection with the connecting tubing. The connection of the indicator to the exhaust port of a 3-71 General Motors engine is shown on Figure 11. To avoid soot from the exhaust getting in the manometer tubes, a thin rubber membrane is interposed between the exhaust and the cover plate of the rotary valve as visible on Figure 11. The manometer tubes corresponding to 0, 10, 20, etc. degrees of crank rotation are identified by a rotatable band type degree scale (Figure 11) near the top of the manometer tubes, which can be adjusted to proper phase relation with the crankshaft.

The rotor is lap fitted in the stator and cover plate which insures a good seal and makes leakage loss negligible. Since mercury is too heavy for accurate reading of exhaust pressures, while water is too light, bromoform (specific gravity 2.87) has been used, with a drop of methyl orange added for better visibility.

Figure 13 is a sample record obtained with the instrument which shows fairly good exhaust conditions for No. 1 cylinder, less good for No. 2 and poorest exhaust for No. 3 cylinder. The



PRESSURE FLUCTUATIONS IN THE EXHAUST DUCTS OF A THREE CYLINDER, TWO STROKE CYCLE DIESEL ENGINE. ENGINE FIRING ORDER 1-3-2. SPEED 1200 R.P.M. LOAD: 35 P.S.I. B.M.E.P. EXHAUST TEMPERATURE 500°F.

FIGURE 13.—EXHAUST PRESSURES OF A G.M. 3-71 TWO-STROKE CYCLE ENGINE FOR EACH OF THE THREE CYLINDERS.

exhaust temperature reading was highest for No. 3 and lowest for No. 1 cylinder.

In examining the exhaust pressure records of multicylinder engines, it must be kept in mind that with a common exhaust header or manifold it is seldom possible to create perfect tuning for each cylinder, as the exhaust impulses necessarily intermingle and the exhaust of one cylinder even "backfires" into the open exhaust of another cylinder which succeeds the former in firing order. Various schemes are being used to minimize the effect of undesirable pressure fluctuations in the common header of multicylinder engines, by properly placed mufflers and silencers, tapered exhaust nipples and other arrangements calculated to dissipate the pressure energy of the exhaust, and prevent it from interfering with the scavenging process. The exhaust conditions in a multicylinder engine with a common exhaust header are, therefore, never ideal. We must be satisfied if the exhaust pressure indicator shows the absence of high instantaneous pressures in the exhaust ducts during the respective scavenging period.

In order to effectively tune a multicylinder engine, individual exhaust pipes of tuned lengths must be mounted between the individual cylinders and the common header or exhaust pot. As an alternative, two or three cylinders may exhaust into a single exhaust pipe, provided the firing order is such that the exhaust periods of the cylinders exhausting into a common pipe do not overlap.

In conclusion, it may be in order to point out that marine applications offer a highly fruitful field for applying exhaust tuning. Auxiliary units for generating electrical power, whether AC or DC, are run at constant speed. Some main propulsion engines have electrical drives. For engines mechanically connected to the propeller, the power requirement varies approximately as the cube of the rotating speed and sharp tuning for maximum speed is desirable, because that is the only occasion when maximum power is needed.

Any two-stroke cycle engine, single or multicylinder, with a properly tuned exhaust system, delivers more power than the same engine would if exhausting into the open with zero length

exhaust pipes. Unfortunately, most engines in service deliver considerably less power than they would with zero exhaust pipes. Neutralizing the interference of improperly tuned pressure waves increases the power output and reduces the exhaust temperature of some cylinders and of the entire engine. This can be accomplished by suitable exhaust ducts, large exhaust headers, and various damping devices. Best results, however, are obtained by a correctly tuned exhaust system when the pressure waves actively assist in scavenging and cylinder charging.

CONCLUSIONS.

1. By proper tuning of the exhaust the performance of two-stroke cycle engines can be appreciably improved.

2. The use of large exhaust pipe diameters and high scavenge pressures are often not justified except on the basis that they tend to minimize the effects of the "incalculable" pressure waves.

3. For many installations the natural frequency of the exhaust system can be calculated and the optimum pipe lengths predicted from design data.

4. Multicylinder engines with common exhaust headers or manifolds also can be improved by proper exhaust tuning but no mathematics is available to handle the problem.

5. With the use of an exhaust pressure indicator, the tuning of any engine may be improved. A recently developed rotary valve indicator should greatly facilitate experimental exhaust tuning.

6. For best tuning, a multicylinder engine should have individual exhaust pipes for each cylinder, or for each two or three cylinders whose exhaust periods do not overlap.

7. If correct tuning of the exhaust system is out of reach, the neutralizing of the interference of improperly timed pressure waves usually improves engine performance.

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PRESSURE-WEAR INDEX: A MEASUREMENT OF EXTREME-PRESSURE PROPERTIES OF LUBRICANTS. (1)

By Robert C. Adams, Sidney M. Collegeman and William C. Clinton.*

The continuing trend to higher power/weight ratios has led to increases in unit pressures in gear design which, in many instances, have exceeded the lubricating properties of straight petroleum materials. This inability of the unfortified materials to lubricate satisfactorily may be the result of imperfection in mechanical design or construction. Ordinary lubricants have proved incapable in preventing the galling and seizing of power-transfer surfaces in existing gear mechanisms. The need for anti-welding, so-called extreme-pressure lubricants, presents the concurrent need for methods and devices for the measurement of extreme-pressure properties.

The four machines which have been most widely used are the SAE, the Timken, the Almen and the Falex. The authors have used the first two of these. The results with both are influenced by the finish on the contact surfaces. Usually, the load at seizure increases with the roughness of the surfaces, but this is not true for all additives. Consistency of measurements, therefore, demands very close control of the surface finish, apparently a closer control than can be obtained without inordinate rejection or reworking of test cups. A more serious objection, at least in the case of the SAE machine, is that a numerical measure of the most potent extreme-pressure lubricants cannot be obtained since the capacity of such lubricants exceeds that of the machine. The machine serves only as a *go, no-go* gauge instead of providing a complete scale of measurement.

The Four-Ball Machine first reached the attention of the

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(1)—This paper has been revised and amplified from one of the same title read to the National Lubricating Grease Institute, November 1943.

*Respectively, Assistant Head of Chemical Laboratory, Assistant Chemist and Junior Chemist at the U. S. Naval Engineering Experiment Station, Annapolis, Md.

authors in 1939(2). In this device one steel ball is rotated about its vertical diameter, while being supported on the points of contact with three similar balls clamped into a cup filled with the lubricant to be tested. During this rotation any predetermined axial load can be applied by weighting a lever which forces the lower balls upward against the rotating ball. Theoretically, the axial load can be increased until either the balls or the lubricating film ruptures, so that there should be no upper limit of measurement. At that time, Four-Ball Machines were available only from the designers in The Netherlands. Several were imported and the original design was approximated in American laboratories, but the Station had to await domestic production in 1941(3).

The domestic Four-Ball Machine is illustrated in Figure 1. The shaft of the 1800 Rpm. vertical motor has a ball chuck at the lower end. The ball pot, shown in the exploded view of Figure 2, holds three balls and the test lubricant. It is supported on a thrust bearing at the short end of the load-lever arm. The torque arm of the ball pot, which serves as a convenient handle in assembling, is restrained from rotation by the calibrated spring to which it is connected by a flexible arm. The extent of compression of the calibrated spring is recorded by a stylus on sensitized paper stretched around a rotating drum. When the spring and the effective length of the torque arm are selected in accordance with the scale of the machine, normal movement of the stylus will not extend to the full width of the paper except on the occurrence of seizure. The drum holding this paper is driven by a synchronous motor, which can be started simultaneously with the large vertical motor.

Several methods for utilization of the Four-Ball Machine in measurement of extreme-pressure properties have been proposed by various investigators. Most of these are based upon use of the characteristic wear-load diagram, which is obtained by plotting on logarithmic coordinates the diameter of the wear spots on the supporting balls against the axial load. Figure 3 is a

(2)—*The Use of the Four-Ball Apparatus for Predicting the Service Performance of Extreme-Pressure and Hypoid Lubricants* by F. L. Garton—Informal Symposium on Lubricants, Technical Committee B, A.S.T.M. Committee D-2, Atlantic City, 28 June 1939.

(3)—Available from Precision Scientific Company, Chicago, Ill.

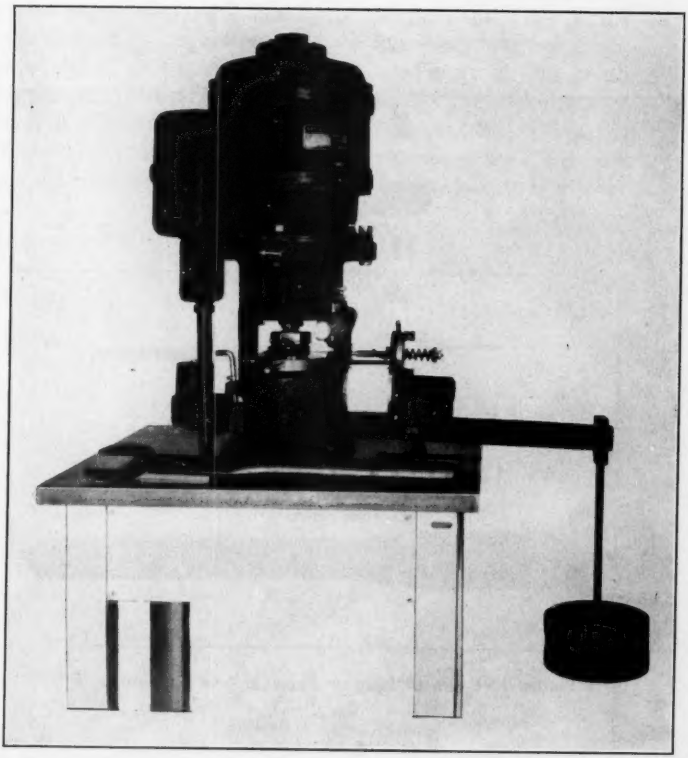

FIGURE 1.—FOUR-BALL MACHINE FOR TESTING EXTREME-PRESSURE LUBRICANTS.

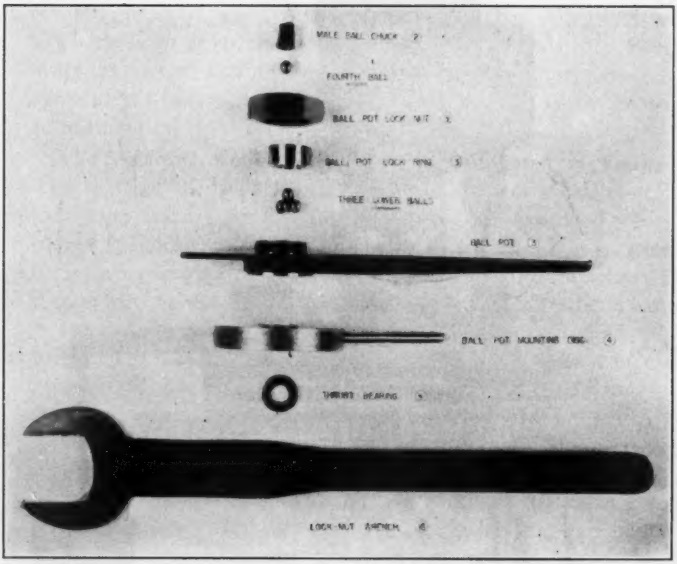

FIGURE 2.—PARTS IN LINE OF PRESSURE AND ROTATION.

somewhat idealized wear-load diagram. Figure 3 also shows the Hertz line of elastic deformation (4) which represents the diameter of the plane of contact between the upper ball and each of the lower balls as a result of static pressure. With a perfect lubricant the wear-load diagram would follow the Hertz line; any divergence above this line is an indication of wear of the balls permitted by imperfection of the lubricant. At low loads, as represented in the first limb, AB, of Figure 3, the graph diverges only slightly from the Hertz line. As the load increases, Point B is reached at which the graph breaks sharply upward from the Hertz line. At still higher loads, where seizure occurs instantaneously on beginning rotation under load, there is another

(4)—A mathematical discussion of elastic deformation is given in the instruction booklet accompanying the Four-Ball Machine (8).

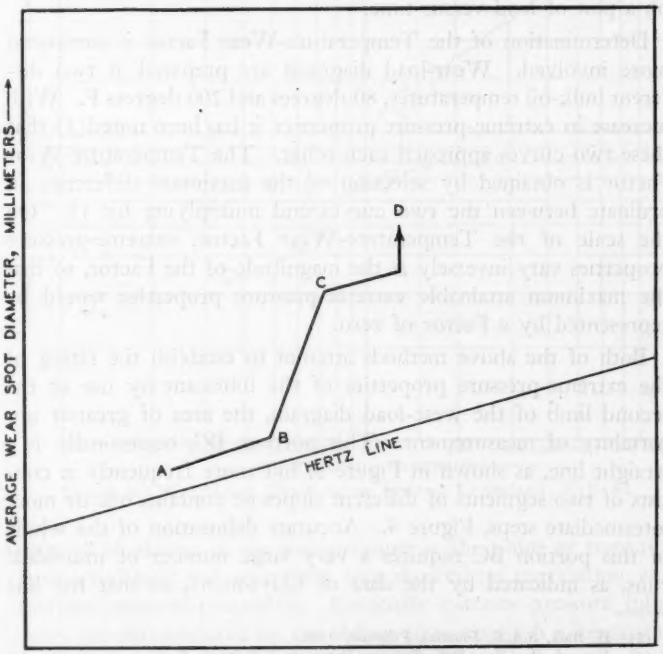

FIGURE 3.—WEAR-LOAD DIAGRAM.

point of inflection, Point C, and the third limb may be approximately parallel to the first. The first limb, AB, is characterized by no seizure of the bearing surfaces; the second limb, BC, is the region of delayed seizure; and in the third limb, CD, immediate seizure occurs. At Point D, which usually is not determined, actual welding of the balls occurs and the curve starts vertically upward.

The seizure-delay method consists of the determination of the maximum load which the lubricant can support for 2-1/2 seconds before seizure. It is determined by making a series of one-minute runs with loads at which delayed seizure occurs and measuring the time from the start of each run to seizure. Since direct measurement of the 2-1/2 second-delay load would be fortuitous, the value is found by interpolation or extrapolation on a plot of load versus time.

Determination of the Temperature-Wear Factor is somewhat more involved. Wear-load diagrams are prepared at two different bulk-oil temperatures, 80 degrees and 200 degrees F. With increase in extreme-pressure properties, it has been noted(5) that these two curves approach each other. The Temperature-Wear Factor is obtained by selection of the maximum difference in ordinate between the two curves and multiplying by 15. On the scale of the Temperature-Wear Factor, extreme-pressure properties vary inversely as the magnitude of the Factor, so that the maximum attainable extreme-pressure properties would be represented by a Factor of zero.

Both of the above methods attempt to establish the rating of the extreme-pressure properties of the lubricant by use of the second limb of the wear-load diagram, the area of greatest uncertainty of measurement. This portion, BC, occasionally is a straight line, as shown in Figure 3, but more frequently it consists of two segments of different slopes or contains one or more intermediate steps, Figure 4. Accurate delineation of the whole of this portion BC requires a very large number of individual runs, as indicated by the data of Clayton(6), so that the line

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(5)—H. Blok, S.A.E. Journal, February 1940.

(6)—*Use of the Four-Ball Extreme-Pressure Lubricant Testing Apparatus for Ordinary Lubricants*—D. Cayton, Inst. Mech. Engrs., General Discussion of Lubrication and Lubricants, October 1937.

can then be located by mathematical reduction of the multiple data.

A first attempt was made to use the terminus of the first limb of the wear-load diagram, Point B, in search for an alternate method for use of the Four-Ball Machine. This would appear reasonable if the criterion of extreme-pressure properties is film strength, the pressure necessary to wipe away the film of lubri-

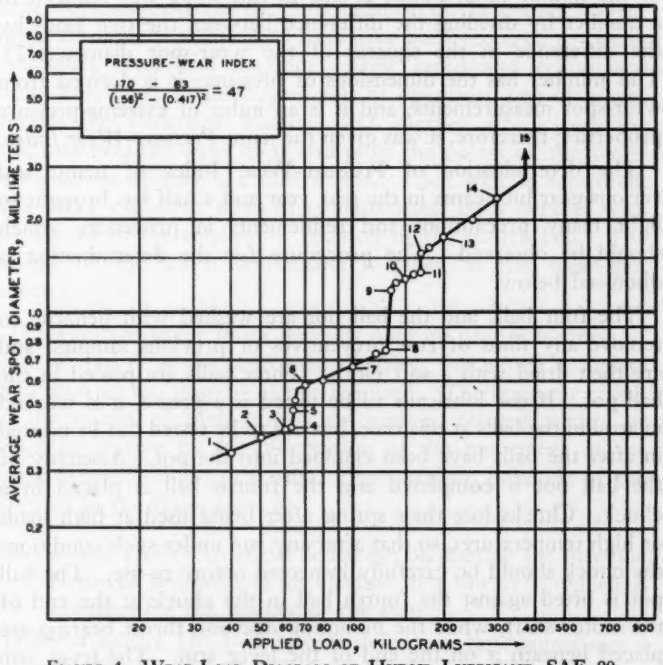

FIGURE 4.—WEAR-LOAD DIAGRAM OF HYPOID LUBRICANT, SAE 80.

cant. Calculation of the unit pressure at the point of transition from axial load and wear spot area showed no correlation with extreme-pressure properties. Evidently extreme-pressure lubricants are characterized by the ability to re-establish a lubricating film after rupture, instead of by unusual resistance to initial rupture.

It was then observed that with straight mineral oils the second limb of the wear-load diagram, BC, was almost vertical, while with increase in extreme-pressure properties or per cent of extreme-pressure additive, this limb rotated clockwise on the end of the first. It appeared, therefore, that the slope of the middle limb could be used as an index of extreme-pressure properties. After determination of the coordinates of the ends of the middle limb, Points B and C, this slope was reduced to a number by dividing the difference between the two loads by the difference in the squares of the wear-spot diameters(7). This number has the dimensions of pressure; it is derived from wear-spot measurements; and it is an index of extreme-pressure properties; therefore, it was given the name *Pressure-Wear Index.*

The determination of Pressure-Wear Index of many and various gear lubricants in the past year and a half has brought to light many precautions and refinements in procedure which should be observed. The procedure for the determination is discussed below.

The four balls and the ball pot are washed with benzene to remove any films of rust preventives or previous samples, and are then dried with a soft cloth. Three balls are placed in the ball pot. If the lubricant to be tested is a grease, it is worked in around the balls at this time, but oils to be tested can be poured in after the balls have been clamped into the pot. Assembly of the ball pot is completed and the fourth ball is placed in a chuck. Chucks lose their spring after being used at high loads or high temperatures, so that after any run under such conditions the chuck should be carefully inspected before re-use. The ball pot is lifted against the fourth ball in the chuck at the end of the motor shaft, while the mounting disc and thrust bearing are placed beneath it on the end of the lever arm. The lever arm is then released and its short end raised carefully against the ball pot assembly to avoid any shock load against the contact surfaces of the balls. After the desired load is placed on the lever arm, the arm is pressed down by hand to insure a firm seating of the top ball on the balls in the cup. Finally, the spring is

(7) $P\text{-}WI = \frac{L_c - L_b}{D_c^2 - D_b^2}$; where L = load, kg., and D = diameter, mm.

connected to the torque arm at the point corresponding to the load on the lever arm. The motor is switched on, run for exactly one minute, and turned off. The load is removed, the lever arm raised and locked, and the ball pot removed and disassembled. The three balls from the ball pot are washed with benzene and retained for measurement while the fourth ball is discarded. Usually between ten and twenty such runs, at different loads, are required for determination of the Pressure-Wear Index, although as many as 70 runs have been required to obtain a complete wear-load diagram. Most rapid and economical results have been obtained by a pair of operators, one of whom operates the Four-Ball Machine and another who cleans the balls, measures the wear spots and plots the points on the wear-load diagram as they are determined. Each spot is measured, with either a filar or eye-piece micrometer, both parallel with and normal to the equator of rotation of the fourth ball. The average of the six measurements is taken as the wear-spot diameter.

The Pressure-Wear Index depends only on proper location of Points B and C of the wear-load diagram, for which complete delineation of the diagram is not required. The operator classifies the results of each run as *no seizure*, *delayed seizure* or *immediate seizure* by his observation of the movement of the torque indicator during the run. Characteristic behavior of the indicator in the three cases is:

No seizure (AB)—An initial deflection on starting the machine followed immediately by decline to a low level which is constant for the balance of the run;

Delayed seizure (BC)—A second deflection (and sometimes a third or fourth) after the initial deflection, followed either by decline to the previous low level or continuance at approximately the maximum level of the deflection;

Immediate seizure (CD)—The initial deflection extends to the full limit of movement of the indicator where it may remain as long as ten seconds before declining to the low level at which it remains, without secondary deflections, for the balance of the run.

In the vicinity of the transitional points, B and C, the information supplied by the torque indicator must be supplemented to obtain precise location of these two points. Positive identification of Point B is obtained by microscopic examination of the wear spot. If no seizure has occurred during the run, the spots are polished circles, less than 0.5 mm. in diameter, usually without pits or scratches within the contact area. If there has been any galling or seizure during the run, the wear spots are marred by pits and striations along the direction of rotation of the fourth ball. These striations extend through the circumference of the wear spot. This distinction is illustrated in Plates 3, 4 and 5 of Figure 5. The highest load at which no seizure occurs provides Point B.

Location of Point C is more difficult. Runs should be made to provide at least three points on the line of immediate seizure (CD) to establish its slope and the runs then extended to lower loads in search for the terminus. Below Point C, in the region of delayed seizure, points on the wear-load diagram tend to scatter, probably as a result of vibration set up by partial seizure and recovery. In the vicinity of the suspected Point C, it is desirable to make triplicate runs, at loads close together, and then plot the average of the 18 wear-spot measurements at each load. If the average of one of the three sets of measurements differs from the grand average by more than five per cent, that set should be excluded from the average plotted in the wear-load diagram. An exact location of Point C has always been obtained graphically by this method, but Figure 4 illustrates how an erroneous value could be obtained if the region between Points 9 and 12 had not been explored.

Figures 4 and 5 illustrate the inter-relation of the wear-load diagram and the microscope in determination of the Pressure-Wear Index. The entire diagram was prepared as an example although runs at loads between 65 and 150 kilograms might have been omitted for determination of the Pressure-Wear Index. The progressive increase in size of spots and in severity of scoring of the contact areas is apparent. Plate 16 shows the rolled edge which surrounds wear spots after severe seizure. This edge must be removed with a sharp blade before spot diameters can

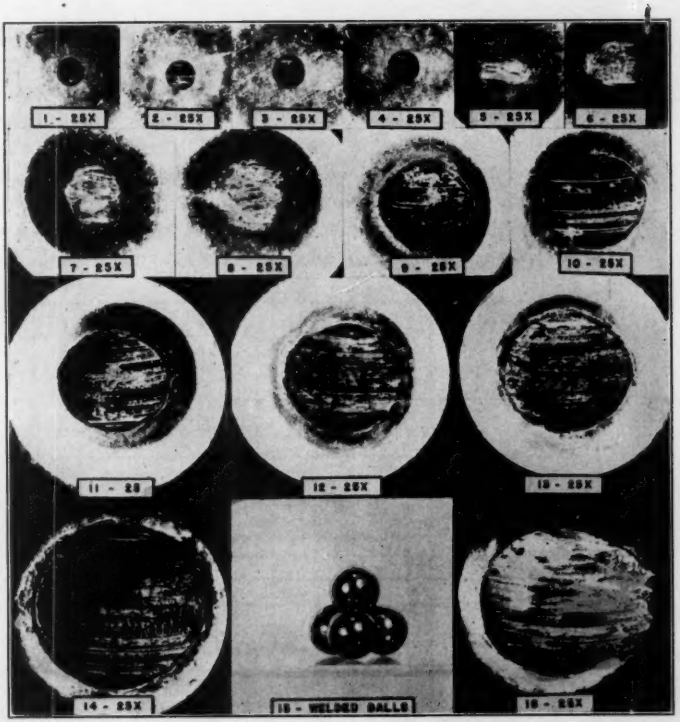

FIGURE 5.—PHOTOGRAPHS OF WEAR SPOTS FROM P-W I DETERMINATION OF FIGURE 4.

be measured with accuracy. Plate 15 shows the four balls welded together as occurs at Point D and higher loads.

The runs comprising a determination of Pressure-Wear Index should not be interrupted but should be completed within a single day. Although separate determinations made at different times on the same lubricant are in satisfactory agreement, some displacement of the entire curve has been observed. The source of this variation has not been located, although both the vibration of the machine support and atmospheric temperature and humidity have been suspected.

The preceding instructions will suffice for determinations on over 95 per cent of the lubricants on which it is attempted. Occasionally, however, an unusual material will yield inconsistent or anomalous results. One such case is illustrated in Figure 6.

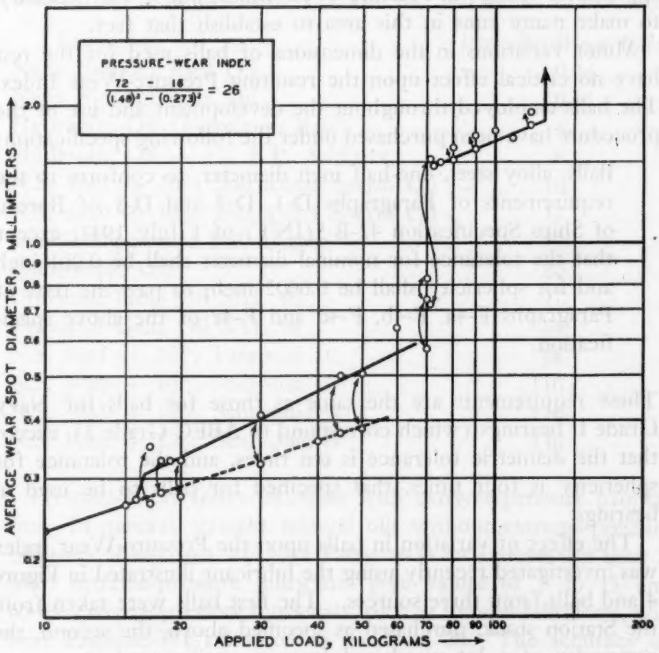

FIGURE 6.—WEAR-LOAD DIAGRAM SHOWING DOUBLE BRANCHES.

This lubricant gave an occasional delayed-seizure, as evidenced by smeared wear-spots, at the same loads at which no-seizure also was obtained. Plotting of all points yielded the two converging lines shown, but it was necessary to reduce the test load to 16 kg. before consistent wear-spots without seizure could be obtained each time. The intersection with the 16 kg. abscissa was taken as Point B.

In the case illustrated in Figure 6, limb CD of the wear-load diagram also showed double seizure. Immediate seizure occurred at the loads in this range, but in some cases a second seizure, as shown by the torque indicator, occurred within the sixty seconds of the run. The double-seizure and immediate single-seizure points traced two lines on the wear-load diagram. The lower terminus of either line could be used as Point C without appreciable change in Pressure-Wear Index, but it was necessary to make many runs in this area to establish that fact.

Minor variations in the dimensions of balls used for the test have no critical effect upon the resulting Pressure-Wear Index. The balls employed throughout the development and use of the procedure have been purchased under the following specification:

> Balls, alloy steel, one-half inch diameter, to conform to the requirements of Paragraphs D-1, D-2 and D-5 of Bureau of Ships Specification 42-B-5(INT) of 1 July 1941, except that the tolerance for nominal diameter shall be 0.001 inch and for sphericity shall be 0.0002 inch; to pass the tests of Paragraphs F-4a, F-4b, F-4c and F-4e of the above specification.

These requirements are the same as those for balls for Navy Grade II bearings (which correspond to ABEC Grade 3), except that the diametric tolerance is ten times, and the tolerance for sphericity is four times, that specified for balls to be used in bearings.

The effect of variation in balls upon the Pressure-Wear Index was investigated recently using the lubricant illustrated in Figure 4 and balls from three sources. The first balls were taken from the Station stock, purchased as specified above; the second, the remainder of a lot purchased from a small manufacturer and

judged unsuitable by two other owners of Four-Ball Machines; and the third were first quality (Gold Seal) balls supplied by the manufacturer of the Machine. All Pressure-Wear Indexes determined with these various balls lay between 46 and 50. The second lot of balls, while dimensionally almost as accurate as the Gold Seal balls, were slightly harder than either of the others. As a result, the wear-spot diameters obtained with them were less than with the others at all loads, but the Pressure-Wear Index was not affected.

Pressure-Wear Index has been used for evaluation of extreme-pressure properties of a variety of liquid and plastic lubricants. Table I gives representative results from the determination on a variety of well-known lubricants.

TABLE I.—PRESSURE-WEAR INDEXES OF SPECIMEN LUBRICANTS.

Lubricant	*Pressure-Wear Index*
Navy Symbol 3080 Oil	0.36
Navy Symbol 3065 Oil	0.61
Hydraulic Oil (Petroleum Base)	4.
Hydraulic Oil + 1% E.P. Additive	6.
N.S. 3065 + 10% Sulfurized Sperm Oil	10
N.S. 3080 + 10% Sulfurized Sperm Oil	12
Low Temperature E.P. Grease	19
Chlorinated Oil	20
Universal Gear Oil SAE 80	47
Universal Gear Oil SAE 90	69
N.S. 3065 + 10% Parapoid 10	98
N.S. 3080 + 10% Parapoid 10	110
N.S. 3065 + 10% Santopoid S	120
N.S. 3080 + 10% Santopoid S	130

Pressure-Wear Index increases with extreme-pressure properties. In general, straight mineral oils without extreme-pressure additives of any sort will have Pressure-Wear Indexes less than 2. Mild extreme-pressure lubricants give values of 5 to 15 and active extreme-pressure lubricants, such as hypoid lubricants, will have Pressure-Wear Indexes in excess of 20. The accuracy of the determination decreases at the higher values, so that Pressure-

Wear Indexes above about 75 are not precise, although values as high as 1100 have been measured. Present accuracy of the method does not justify reporting determined values to more than two significant figures.

The Pressure-Wear Index appears to have several distinguishing merits. First, the expendable test parts, the balls, are cheap, virtually identical and are readily available from several sources. The availability at the present time is, of course, conditioned by priorities, but in peace time an almost unlimited supply is readily available at a price of about one cent each or four cents per run. Second, the method assigns a numerical value to the extreme-pressure properties of any lubricant; no lubricant is so potent as to be beyond the scale of measurement. Finally, there is an internal check on the reliability of results, since the determination comprises a succession of individual measurements, so that erratic points are apparent. With some materials, difficult to evaluate, this makes the determination tedious although enlightening.

The pressure of circumstances forced the Navy to employ the Pressure-Wear Index after less study and trial than normally would have been desirable. The procedure can scarcely be considered to have passed the development stage when so few laboratories have used it. At the present time round-robin tests among the owners of Four-Ball Machines are in progress. Upon the completion of these tests, it is hoped that the method will be more widely appreciated. The significance of the Index still awaits delineation by results of service tests, the ultimate criterion. Despite all these limitations, the Pressure-Wear Index appears to be a reproducible and useful measure of extreme-pressure properties, so that the Navy probably will continue its use and will welcome its investigation and criticism by others.

REFLECTIONS ON RECENT PUBLICATIONS ON GAS TURBINE DEVELOPMENTS.

By Frederick Nettel.

Two papers were presented recently in quick succession on Gas Turbines*, a clear indication that American industry does not want to be taken by surprise by a development which was slow in coming, but which is beginning to stir the engineering field all over the world.

Mr. Moss devoted the major part of his paper to historic reminiscences which are fascinating, showing as they do the tremendous handicaps under which the early workers had to labor. While the primary gas turbine research work in Mr. Moss' organization ceased in 1907 (it seems to have been revived more recently), work was going on abroad with varying intensity and still more varying success.

Mr. Moss' statement about Holzwarth "there seems doubt as to whether he ever reported performance of a complete machine which compressed its own air", is more or less confirmed by the late Prof. Stodola. Holzwarth appears primarily as designer who had to be taught by decades of strenuous work that the gas turbine is more than a problem of temperatures, and that a little more attention to thermodynamics would have saved him and his backers many a headache and expense. Thus, in spite of the impetus which Holzwarth's untiring efforts have given to the gas turbine problem, the neglect of the thermodynamic aspect retarded promising proposals from other sources.

In Europe it was in the meantime recognized that, if the gas turbine should ever be put on the map, an efficient compressor was needed. Centrifugal compressors were good by that time, but by far not good enough; efficiencies above 82 or possibly as

*"Gas Turbines and Turbosuperchargers" by S. A. Moss, read before the A.S.M.E. Meeting 1943 at New York. "The Gas Turbine as a Possible Marine Prime Mover" by Prof. C. R. Soderberg and R. B. Smith, read before The Society of Naval Architects and Marine Engineers at 1943 Meeting.

high as 86 were needed. Sir Charles A. Parsons was amongst the first to see a possibility in the axial flow type compressor but he had to fail since at his time nothing was known about airfoil performance. Advantage was promptly taken by European engineers to apply the lessons taught by aerodynamics to compressor design, and the results achieved fully justified the efforts. Efficiencies up to 84.5 per cent, referred to the "adiabatic," have been measured for a large compressor having a pressure ratio of about 4.35. Thus at last that part of the gas turbine problem appeared to be solved in an absolutely reliable and safe manner, involving no element unknown to the steam turbine designer.

In this connection attention should be drawn to efforts to build efficient positive displacement compressors. The basic idea underlying this work was to avoid the well-known tendency of the centrifugal and axial flow compressor to develop an instability of flow (also called "pumping" or "surging"), if the working conditions deviate considerably from the design condition. The outcome is a screw compressor on the principle first employed by Krigar in 1878, perfected to a remarkable degree by close study of the air flow in, through, and out of the compressor, and by application of close tolerances in manufacture made possible by modern machine tool practice. While the efficiencies stated by the makers as obtainable are good for the output and speed ranges which are of interest for gas turbine plants, there is no indication as yet of any superiority over the axial flow compressor. Moreover, the latter allows the use of larger clearances, and does not involve a gear as the positive displacement type does.

As to the dreaded instability of the axial flow compressor, considerable headway has been made in reducing that danger by proper hook-up (multi-shaft arrangements) and proper methods of load regulation. For special purposes arrangements have been found in which the instability can be eliminated altogether.

It must, however, be recognized that, in spite of the above critical remarks, the modernized screw compressor is a valuable contribution to gas turbine development. Its effect will be felt also in other fields in future. The argument: axial flow compressor vs. positive displacement compressor is bound to con-

tinue for some time, fanned more by conflicting commercial and patent interests than by vital engineering considerations. So far, the axial flow type appears in front.

After a turbine (expansion) efficiency of 88.4 per cent had been measured by Stodola for a turbine operating at an expansion ratio of about 4.3 and a gas flow of about 500,000 pounds per hour, such efficiencies, and even higher ones, are being assumed in various quarters for much smaller units. Soderberg-Smith, for example, assume an internal efficiency of 90.8 per cent for "units in excess of 1500 Hp." allowing 2 per cent for mechanical losses. (The latter figures must appear rather high for large turbines!) These and other published figures should advise caution when predicting turbine efficiencies which are influenced not only by size but also by the respective expansion ratio.

Thus it will be realized that it is hardly permissible to assume one and the same efficiency figure for all kinds and sizes of turbines as they may be employed in the great number of possible hook-ups.

What is true of turbine efficiencies is even more so for compressor efficiencies. Here we are confronted with a problem of special significance: Practically all publications on gas turbines involving cycle calculations start from the assumption of internal turbine and compressor efficiencies being constant and unaffected by variations in the cycle. This, however, is by no means the case. The internal efficiency of a compressor, for example, changes materially with its pressure ratio and with its intake volume. Both vary when the cycle data are varied.

More generally it can be said that each row of blades in a turbine or compressor has its local efficiency dependent on its size; local losses are caused in each stage. With this local internal efficiency the turbine or compressor blading works on a medium whose temperature depends on the local losses in the preceding stages. These two effects, namely the local loss and the "interheating effect" of the preceding stages as a result of losses in these stages, act additively in a compressor and subtractively in a turbine.

The size (length) of the blading at a given point depends on the output of the plant, the air rate, and the location of said

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The size (length) of the blading at a given point depends on the output of the plant, the air rate, and the location of said

point in the cycle. The two latter vary with every variation of the contemplated cycle, and so does the amount of the mentioned interheating effect.

These somewhat involved relationships seem to have so far escaped the attention of the investigators of gas turbine cycles; they appear replaced by the simple assumption of constant internal efficiencies. Even the higher pressure stages in one and the same multi-stage cycle were, in some papers, assumed to have the same efficiency as the lowest pressure stage, both on the turbine and the compressor side. However, by maintaining this illusory assumption over wide ranges of hook-ups, the picture of their absolute and relative merits emerges distorted.

An actually less favorable cycle may appear as the best one for given conditions, and for one particular cycle the hook-up and the cycle data, such as pressure ratios, may be chosen not at their best, due to this assumption of constant internal efficiencies.

What is worse, and liable to impair a sound development of the gas turbine, is the fact that the neglect of the mentioned involved efficiency relationships blurs the proper recognition of the economic lower output limit for each cycle and hook-up. Far too elaborate cycles are already being contemplated for too small outputs. If these plants were built, disappointment is bound to follow, as the very small top compressors of such multi-stage cycles cannot live up to the internal efficiency values assumed.

In order to illustrate the practical effect of these efficiency considerations, let us take Figure 11 of the Soderberg-Smith paper. A 3-3-0, 75 cycle at 1200 degrees F., when built for smallest air rate, would have a total pressure ratio between 18 and 20. Its last compressor, however, would even in a plant of 3000 Hp. output have a discharge volume of about 1000 cubic feet per minute. Such compressor, irrespective of type, axial flow or positive displacement, can hardly be expected to reach even 80 per cent efficiency, not to speak of 84.7 per cent.

If we further bear in mind that every per cent drop in compressor efficiency affects the cycle efficiency by a multiple of one per cent, it becomes obvious that greatest caution is indi-

cated in the interpretation of published curves, lest faulty conclusions be arrived at, not only relative to actually obtainable cycle efficiencies, but also relative to the lower economic output limit for every individual hook-up.

Computations on the basis of varying efficiencies show that even the comparatively simple hook-up 2-1-η_r (according to the Soderberg-Smith nomenclature), involving one intercooler, one reheater, and a heat exchanger, does not make it advisable for application for outputs below 3500 Hp. More elaborate cycles require much larger outputs.

Developments in the gas turbine field during the last five years or so have followed a rather peculiar trend: It would have appeared logical that, with a highly efficient compressor fully developed, the European manufacturers should have attempted to get out of the then available cycle hook-ups as much as possible by going deeper into the thermodynamics of the problem. Instead, the leading manufacturer preferred to emphasize the simplicity of the ordinary Brayton cycle, its independence of sources of cooling water, etc., and to predict improvement of the rather mediocre efficiency, if and when metallurgical progress would permit the employment of temperatures in excess of 1000 degrees F. The U. S. licensee of this group appears to have followed the same policy until quite recently.

Another European group sponsored the positive displacement compressor mentioned before. At the same time it undertook a rather comprehensive study of many hook-ups and the load regulation problems involved. Application of the gas turbine for stationary plants, ships, locomotives and even airplanes is envisaged. While these studies are interesting from many angles, they did not bring out any new basic thermodynamic improvement.

Workers in various other parts of Europe contributed meanwhile a great deal to the theory of multiple shaft hook-ups, and in doing so managed to get somewhat closer to a more general conception of the gas turbine as a thermodynamic machine.

Due to the war and other considerations much of the fundamental work in this direction will have to remain unpublished for some time. However, it can be said that the thermodynamic

problem involved can never be satisfactorily solved by trial and error calculations, as have been done so far by all authors. There are too many independent variables involved to do that, even if several hundred thousand cycles would or could be calculated. It is due to this inherent difficulty that curves purporting to show what can be done with the gas turbine must be subjected to further close analysis before being accepted as basis for practical plants.

Another important question is raised in the Soderberg-Smith paper: What is the highest output for which gas turbine plants should or could be built today? These authors state: "Until the promises of high-temperature design have been verified by operating experience it is not to be expected that gas turbines exceeding 7500 Hp. will be constructed." From that paper it is evident that high temperatures beyond reach at present are those above 1200 degrees F. While this temperature limitation must be considered sound at this time, the statement about the possible or recommendable size of gas turbine plants appears unsupported by facts and thus invites closer scrutiny.

That much higher outputs can be reached now, without exceeding 1200 degrees F., and without using compressors or compressor stages not fully developed, will become clear from the following: If we incorporate the Neuchatel axial flow compressor in a cycle 2-1-η_r (one intercooler, one reheater) we arrive at an output of about 15,000 Hp. If we go one step further and incorporate a double flow compressor with largest individual stages same as in the Neuchatel compressor, also in the 2-1-η_r cycle, we arrive at 30,000 Hp.! Since no data are available regarding the largest existing positive displacement compressor, similar calculations are not feasible at this moment for plants involving the use of this latter type compressor.

In connection with good partial load performance Soderberg-Smith appear somewhat over-impressed in favor of the positive displacement compressor. Gas turbine plants using axial flow compressors can also be built to give favorable efficiencies at part load. As a matter of fact, the positive displacement compressor has its own disadvantage in that it acts as a closed valve when at standstill.

This brings up the question of selection of hook-up for marine applications which is one of the principal subjects of the Soderberg-Smith paper. The cycle selected as typical for this purpose was no doubt chosen with a view to good partial load efficiency.

In this connection an old truth must be repeated for the readers of this Journal, namely that for the average merchant vessel part-load efficiency is hardly ever of any importance, while the reverse is true of most naval vessels.

In merchant ships it might, therefore, be of advantage to discount part load efficiency if by doing so other advantages might be secured.

Conversely, in naval vessels maximum load efficiency might be sacrificed if by doing so weight and space are saved.

Considering the fundamentally different operating conditions in these two cases, though both falling in the category "Marine Applications," it must remain open to doubt whether any one hook-up can serve best for merchant and naval needs alike!

The war has brought into being several new types of naval craft, the special duties of which will most probably have to be met by specialized propulsion plants.

The general note of optimism as regards the ultimate use of cheap crude (bunker) oil in gas turbines on ships for all purposes seems most certainly justified. Such optimism is, however, hardly in order for the prospects of gas turbines in stationary plants as long as solid fuels (coal, coke, lignite, etc.) cannot be utilized satisfactorily and efficiently. Experiments are under way in Europe, for some years now, to burn pulverized coal under pressure and to separate the solid residue of combustion (ash, slag) before leading the gases into the turbine. Those with experience with pulverized coal combustion in steam boilers will appreciate the immense difficulties in the path to a successful solution by this basically simplest method.

Other methods have been tried—for example, by resorting to various kinds of closed or semi-closed cycles, working at arbitrary pressure levels, with the lower pressure level mostly selected above atmospheric pressure.

While the prospect of keeping the combustion products

entirely out of the air turbine looks promising at first glance, the necessity to reject much heat from these cycles at low temperature levels spoils some of their prospects since huge additional heat exchangers become necessary for that purpose alone. The manufacturers realize themselves that closed cycles offer little attraction on vehicles of any kind. The idea to regulate the load in closed cycles by varying the density of the working fluid (air) is sound, but requires rather complicated governing gear and may be expected to respond rather slowly to sudden large load increases.

That the problem of utilizing solid fuel in combustion turbines remains of paramount importance is evident if we remember that more than 90 per cent of all stationary thermal power plants in the U.S.A. and nearly 100 per cent in many other countries are burning coal. Considerable progress has been made more recently, which gives hope for an early satisfactory solution, probably by avoiding combustion under pressure altogether.

Very interesting views have been put forward by Soderberg-Smith relative to the starting of gas turbine plants, with particular reference to marine applications. European practice utilizing electric motors or auxiliary Diesels is considered awkward, and air bottles charged by the main compressor or by a "small" auxiliary air pump are advocated. Even if, as stated, 35 pounds of air at 80 psi should suffice for a single start of a 2500 Hp. plant, a multiple of that air weight would have to be stored on a ship to take care of "false starts" and emergency conditions. Even a "small" auxiliary air compressor requires a source of power. If we survey the situation carefully, we will see that the use of electric energy for starting, especially for turbo-electric marine installations, can be made very advantageous. On every merchant or naval craft there will be auxiliary generator sets for which Diesel drive is generally accepted as the best, and for which gas turbine drive is hardly suitable due to the relatively small capacities involved. In case of turbo-electric transmission to the propeller, these sets will often be found of sufficient capacity to start the gas turbine plant. These sets may, moreover, if suitably designed, for d.c. or a.c., be used to propel the ship at low speed, for example, in case of trouble

with the gas turbine, and thus keep it maneuverable in an emergency. Hardly any extra weight or space are required.

Mr. Moss deals at some length with arrangements involving the use of reciprocating (Diesel) engines in combination with gas turbines. Such plants can be and have been built, but their existence from an engineering and economical standpoint could be justified only by an overwhelming superiority in efficiency over the pure-turbo gas turbine. Such superiority, even today, is doubtful, and it will finally disappear when top temperatures higher than 1100 degrees F. will be employed.

Amongst the special applications of the gas turbine, the gas turbine locomotive (not necessarily with electric power transmission) and siderurgical blower drives show considerable promise.

The most controversial question in the whole gas turbine problem is: What top temperatures should be permitted now or in the near future? Most of the recent publications give the impression that responsible designers consider 1200 degrees F. within the realm of immediate possibility; some declare readiness to go even higher. Some utterances could, indeed, mislead us into believing that 1500 degrees F. are, so to speak, round the corner.

If we survey the plants which have stood up to more or less continuous service, we find temperatures up to 570 degrees C. or 1060 degrees F. It is true that these plants have been designed and built several years ago so that, allowing for metallurgical progress since then, 1200 degrees F. may appear a reasonable figure. Soderberg-Smith do not share the optimism with regard to uninterrupted advance of operating temperatures for long-time service. They add, very significantly, that the efficiencies obtainable at 1200 degrees F. do not indicate that extremes of temperatures are absolutely essential in order to justify gas turbine developments in the immediate future. Experience with aircraft engine superchargers is of very limited value for the primary gas turbine.

Soderberg-Smith's attitude that "ground (in the use of temperatures in excess of 1200 degrees F.) can be gained only slowly and with caution" can be accepted without reservation.

In its further support it can be stated that, if we use all presently available means thermodynamically correctly, if we avoid wishful thinking with regard to obtainable compressor and turbine efficiencies, if we choose a plant of suitable output at which the rotating machines can work to best advantage, if we select a suitable hook-up which will be different for each purpose, size and required performance, even with a highest temperature of 1050 degrees F., a thermal overall efficiency at rated load, referred to the coupling of the useful power turbine, of thirty per cent is reliably obtainable, provided the plant has a capacity of about 4000 Hp. or preferably more.

No steam plant of that output on land, and still less on board ship, has ever been measured to give such efficiency. Thus a logical question arises: Why at this moment should we "shoot higher," especially if we consider that savings in weight, space, and, for stationary plants, cooling water, already make the gas turbine potentially superior to steam plants?

Caution in raising of temperatures will avoid setbacks which are liable to be unjustly blamed on the gas turbine as such and which are likely to retard rather than assist its healthy growth.

LAMINAR FLOW FRICTION LOSSES THROUGH FITTINGS, BENDS AND VALVES.

By Cyrus Beck.*

NOTE: The opinions or assertions contained in this article are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the Naval service at large.

I—Introduction.

Pressure losses under laminar flow conditions for 3½ inch I.P.S. fittings, bends and valves, together with a suitable although not exact method of combining their equivalent lengths in an assembly were presented in a previous article. It is the purpose of this paper to discuss, using original equivalent length data, the relation of the pressure losses to the character and rate of dissipation of the disturbance immediately after an obstruction such as a fitting, bend or valve.

The problem may be resolved as follows: First, decision as to what quantities and variables adequately describe the effect (friction loss) of a fluid passing through a fitting; second, measurement experimentally of these quantities and variables; third, analysis and interpretation of the observed data to learn how the variables are related.

The work reported in this paper is confined to the laminar flow region covering a Reynolds number range from 50–1000. Commercial seamless, steel tubing 3½ inch I.P.S. (4 inch O.D.) with an average inner diameter of 3.705 inches was used. A globe valve, angle valve, standard 90 degree elbow, long-radius 90 degree elbow, tee, gate valve, and 90 degree circular constant cross-section bends with relative curvatures R/d = 2, 3, and 6 were investigated. The globe valve was tested with flow under and

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over the disc., i.e., normal and reversed positions. The tee was examined for the conditions of straight through flow and branch flow. The valves and fittings were designed for 3½ inch I.P.S. standards while the bends were fabricated from the steel tubing.

II—Method of Procedure.

The test set-up is shown in Figure 1. A straight section of tubing 22.5 feet long was used to establish the friction factor and for the manometer indication per foot of straight tubing utilized in the calculation of equivalent length. The test section for the fittings also was 22.5 feet with a flanged connection 4 feet downstream from the front end. The fittings, valves, and bends tested were installed at the flanged connection to secure flow data immediately after the fitting. Nine piezometer rings were located at distances of 0.5, 1.5, 2.5, 5.0, 7.5, 10.0, 12.5, 15.0, and 18.5 feet downstream from the position of the fitting. These piezometer rings were connected by a manifold to the manometer making it possible to employ any of them for pressure measurements. The upstream piezometer ring for obtaining equivalent length was located 4 feet in front of the fitting.

Two rotating "U" tube manometers with colored carbon tetrachloride as the manometer fluid were used to measure the pressure drops across the straight tubing section and across the fitting. Pressure reservoirs (jars) were employed for transmitting the pressures. An oil layer was maintained at the top of the reservoirs with water on the bottom, the latter medium acting as the connecting fluid between the oil and the carbon tetracholride in the manometer.

An oil preheater was utilized to give the requisite temperature control, although the steam coils also were utilized in the oil storage tanks. Calibrated thermometers measured the entrance and exit oil temperatures. A thermometer, hung on the manometer scale, served to indicate the temperature of the manometer fluid. The fluid used in the system was a Navy Grade fuel oil. The density and kinematic viscosity (by the Fenske modified Ostwald viscosimeter) of the oil were checked frequently. The density-temperature curve of the carbon tetrachloride (saturated with water) was determined before it was introduced into the manometer.

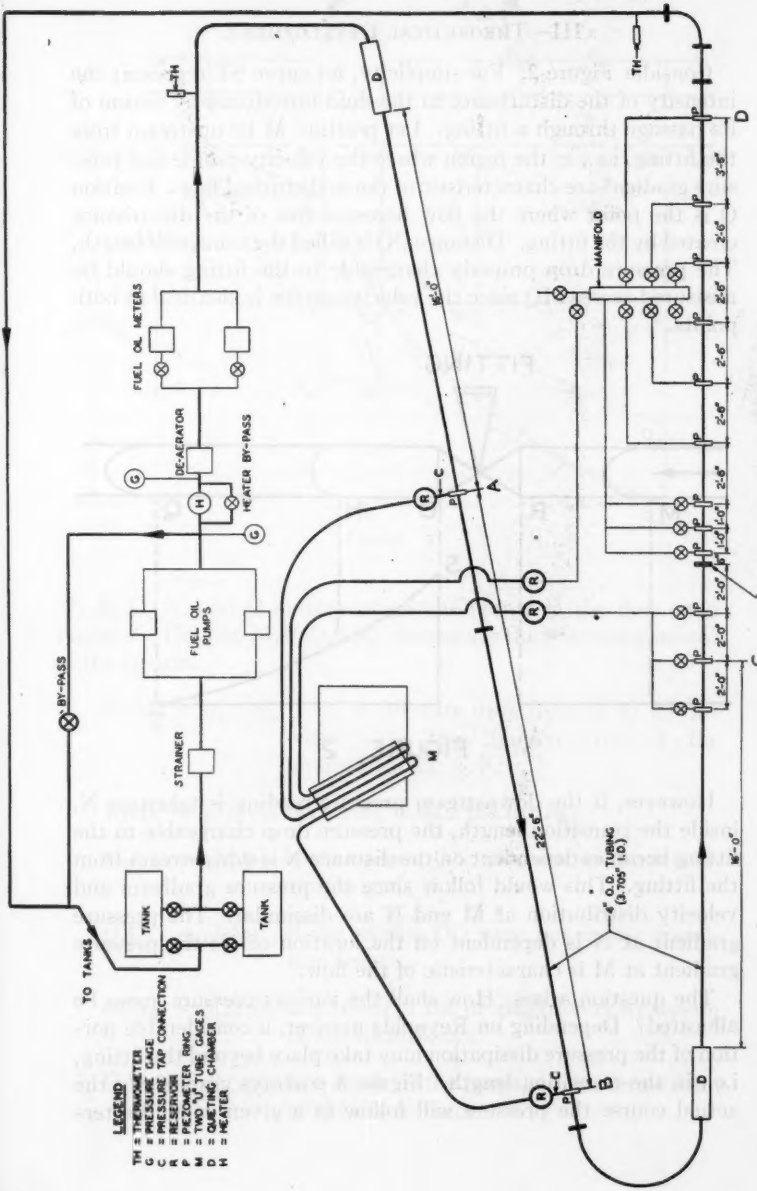

FIGURE I. DIAGRAM OF APPARATUS

III—Theoretical Development.

Consider Figure 2. For simplicity, let curve ST represent the intensity of the disturbance in the fluid introduced by reason of its passage through a fitting. Let position M be upstream from the fitting, i.e., in the region where the velocity profile and pressure gradient are characteristic of the undisturbed flow. Position Q is the point where the flow becomes free of the disturbance created by the fitting. Distance OQ is called the transition length. The pressure drop properly chargeable to the fitting should be measured across MQ since the velocity profile is identical at both points.

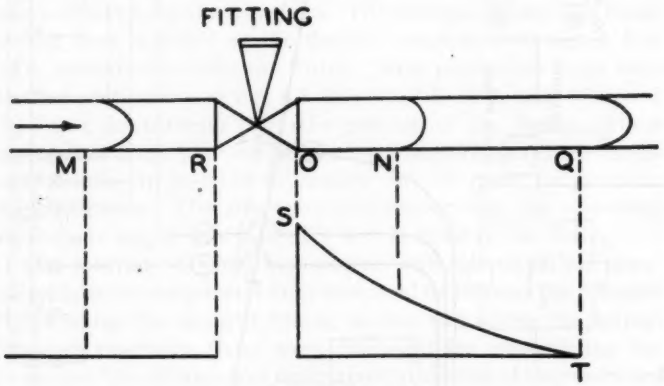

FIGURE 2

However, if the downstream pressure reading is taken at N, inside the transition length, the pressure drop chargeable to the fitting becomes dependent on the distance N is downstream from the fitting. This would follow since the pressure gradients and velocity distribution at M and N are dissimilar. The pressure gradient at N is dependent on the location of N; the pressure gradient at M is characteristic of the flow.

The question arises: How shall the various pressure losses be allocated? Depending on Reynolds number, a considerable portion of the pressure dissipation may take place beyond the fitting, i.e., in the transition length. Figure 3 portrays graphically the actual course the pressure will follow in a given case. Letters

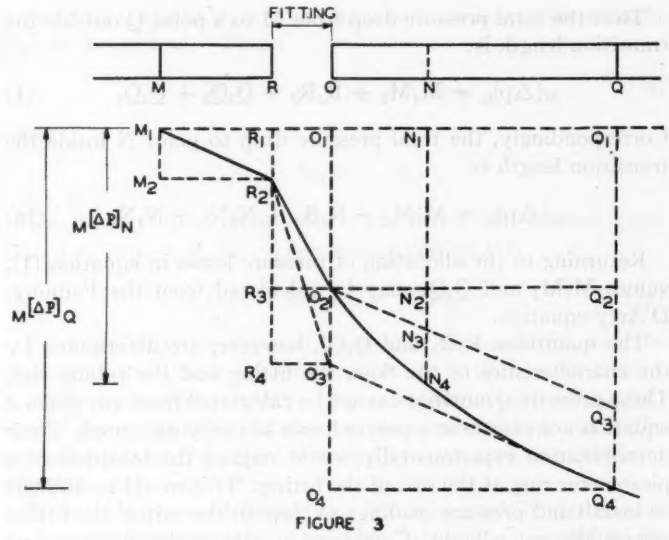

FIGURE 3

M, R, O, N, and Q indicate the same points in the flow as in Figure 2. The line $M_1R_2O_2N_4Q_4$ represents the pressure gradient of the system.

$M_1M_2 = {}_M[\Delta p]_R$ (read as pressure drop from M to R) due entirely to the characteristics of the straight tubing flow.

$R_2R_3 = {}_R[\Delta p]_O$ occurring within the fitting.

$N_2N_3 = {}_o[\Delta p]_N$ if the section ON were not in the transition length.

$Q_2Q_3 = {}_o[\Delta p]_Q$ if the section OQ were not in the transition length.

$N_3N_4 = {}_o[\Delta p]_N$ as a result of the propagation downstream of the disturbance.

$Q_3Q_4 = {}_o[\Delta p]_Q$ as a result of the propagation downstream of the disturbance.

Thus the total pressure drop from M to a point Q outside the transition length is:

$$_{M}[\Delta p]_{Q} = M_1M_2 + R_2R_3 + Q_2Q_3 + Q_3Q_4 \qquad (1)$$

Correspondingly, the total pressure drop to point N inside the transition length is:

$$_{M}[\Delta p]_{N} = M_1M_2 + R_2R_3 + N_2N_3 + N_3N_4 \qquad (1a)$$

Returning to the allocation of pressure losses in equation (1), values M_1M_2 and Q_2Q_3 may be calculated from the Fanning-D'Arcy equation.

The quantities R_2R_3 and Q_3Q_4, however, are determined by the characteristics of the flow, the fitting and the tubing size. These pressure quantities cannot be calculated from any general equation nor can these separate losses be easily measured. Their determination experimentally would require the insertion of a piezometer ring at the exit of the fitting. This would be difficult to install and pressure readings so close to the exit of the fitting are usually not reliable. Combination of the individual pressure losses (R_2R_3 occurring in the fitting plus Q_3Q_4 in the transition length) seemed a simple expedient. Since both losses are caused by the fitting, they should be properly chargeable to it even though part of the pressure dissipation takes place in the pipe downstream.

Referring to Figure 3, in which line O_3Q_4 has been drawn parallel to O_2Q_3, it will be seen that this expedient gives rise to a fictitious but useful pressure gradient, $M_1R_2O_3Q_4$.

Substituting:

$$R_2R_4 = R_2R_3 + Q_3Q_4 \qquad (1b)$$

$$O_3O_4 = Q_2Q_3 \qquad (1c)$$

in equation (1)

$$_{M}[\Delta p]_{Q} = M_1M_2 + R_2R_4 + O_3O_4 \qquad (1d)$$

The quantities $_{M}[\Delta p]_{Q}$, M_1M_2 and O_3O_4 can all be determined. Then the pressure drop chargeable to the fitting is:

$$R_2R_4 = {}_{M}[\Delta p]_{Q} - M_1M_2 - O_3O_4 \qquad (1e)$$

But M_1M_2 and O_3O_4 are the pressure drops due to the straight pipe in the system. Then:

$$R_2R_4 = {}_M[\Delta p]_Q - [\Delta p]_p \tag{1f}$$

where

$$[\Delta p]_p = M_1M_2 + O_3O_4$$

The pressure drop chargeable to the fitting will be designated Δp_o, i.e., $\Delta p_o = R_2R_4$. Then from equation (1f)

$$\Delta p_o = {}_M[\Delta p]_Q - [\Delta p]_p \tag{1g}$$

Equation (1g) was obtained for a point Q outside the wake of the disturbance, but a similar equation can be derived for a point N inside the transition length. Hence, equation (1g) holds regardless of the position of the downstream pressure tap and is the general equation for calculating the pressure loss chargeable to a fitting.

With the pressure drop across a fitting defined, it is necessary to determine theoretically, if possible, how Δp_o is related to the other variables in a given system. Dimensional analysis makes it possible to establish a functional relationship. The method of Lord Rayleigh instead of E. Buckingham was used since it is desirable to learn the dependence of Δp_o on the other variables. Both methods are consequences of the π Theorem.

The assumption is now made that five independent variables are involved and that they completely determine the pressure drop across a fitting when the downstream pressure tap lies in the disturbance. This means that Δp_o is a function of the varibles, density (ρ), viscosity (μ), velocity (V), inside pipe diameter (d) and the distance downstream (x) of the pressure tap from the exit end of the fitting.

From the theory of dimensional analysis, two dimensionless groups should appear. Expressing the relationship functionally

$$\Delta p_o = \gamma(\rho, \mu, V, d, x) \tag{2}$$

$$\Delta p_o \sim \rho^a \mu^b V^c d^e x^g \tag{2a}$$

Substituting in relation (2a), the fundamental dimensions of mass (M), length (L), and time (T)

$$MT^{-2}L^{-1} \sim \left(\frac{M}{L^3}\right)^a \left(\frac{M}{LT}\right)^b \left(\frac{L}{T}\right)^c \left(L\right)^e \left(L\right)^g \tag{2b}$$

$$MT^{-2}L^{-1} \sim M^{a+b}\, T^{-b-c}\, L^{-3a-b+c+e+g} \tag{2c}$$

Equating corresponding exponents gives the following simultaneous equations:

$$1 = a + b;\ -2 = -b - c;\ -1 = -3a - b + c + e + g$$

which when solved in terms of e and g give

$$a = e + g + 1;\ b = -e - g;\ c = e + g + 2$$

hence,

$$\Delta p_o \sim \rho^{e+g+1}\, \mu^{-e-g}\, V^{e+g+2}\, d^e\, x^g \tag{2d}$$

Combining gives

$$\Delta p_o \sim \left(\frac{\rho V d}{\mu}\right)^e \left(\frac{\rho V x}{\mu}\right)^g \rho V^2 \tag{2e}$$

Substituting for the dimensionless groups

$$Re = \frac{\rho V d}{\mu} \text{ and } X = \frac{\rho V x}{\mu}$$

$$\Delta p_o = (Re)^e\, (X)^g\, \rho V^2 \tag{2f}$$

Relation (2e) can be written

$$\Delta p_o = \left(\frac{\rho V d}{\mu}\right)^{e+g} \left(\frac{x}{d}\right)^g \rho V^2 \tag{2g}$$

Letting $D = \frac{x}{d}$, relation (2g) becomes

$$\Delta p_o \sim (Re)^{e+g}\, (D)^g\, \rho V^2 \tag{2h}$$

Dropping the exponents and putting relation (2h) in the more general functional form

$$\Delta p_o = \psi 1\ (Re, D)\ \rho V^2 \tag{2i}$$

As was expected two dimensionless groups appear, $Re \left(= \frac{\rho V d}{\mu}\right)$ and $D \left(= \frac{x}{d}\right)$. From the analysis, $X \left(= \frac{\rho V x}{\mu}\right)$ could be used in

place of D but the latter is considered preferable since it has the physical meaning of "diameters downstream" from the fitting.

The equivalent length of a fitting is defined as the ratio of the pressure drop across and chargeable to the fitting Δp_o, to the pressure drop per foot or per diameter of straight pipe.

Let Δp_c be the pressure drop per diameter of pipe and E the equivalent length in diameters. Then

$$E = \frac{\Delta p_o}{\Delta p_c} \tag{3}$$

From the Fanning-D'Arcy equation, the pressure drop per diameter is

$$\Delta p_c = k\, f\, \rho V^2 \tag{3a}$$

where f is the friction factor and k depends on the units. Substituting from equations (2i) and (3a), equation (3) becomes

$$E = \frac{\psi 1\ (Re,\ D)}{f} \tag{3b}$$

If similar units are used in (2i) and (3a), the k in equation (3a) will cancel out leaving equation (3b) dimensionless.

It is known, of course, that the friction factor for straight pipe is related to the Reynolds number. In laminar flow this relation is represented by Poiseuille's Law and in turbulent flow by several different equations, depending upon the investigator. Expressing the fact that the friction factor is a function of Reynolds number

$$f = \psi_2\ (Re) \tag{3c}$$

Substituting equation (3c) in equation (3b)

$$E = \frac{\psi_1\ (Re,\ D)}{\psi_2\ (Re)} \tag{3d}$$

Let a third function, ψ, equal the ratio $\frac{\psi_1}{\psi_2}$ or

$$\psi\ (Re,\ D) = \frac{\psi_1\ (Re,\ D)}{\psi_2\ (Re)} \tag{3e}$$

ITEM NO.	ITEM	UNITS	FORMULA
1	Run Number		
2	Date		
3	Pipe Diameter		
a	Nominal		
b	Actual I.D.	Inches	Micrometer
c	Actual I.D.	Feet	Item 3b ÷ 12
4	Length of Straight Pipe		
a	Entrance length	Feet	
b	Exit length after the fitting	Feet	
c	Exit length after the fitting in diameters		
d	Piezometer ring number		
e	Length of straight pipe	Feet	
5	Oil Constants		
a	Fuel oil number		Naval Boiler and Turbine Laboratory Designation
b	Average inlet temperature	°F.	Calibrated Thermometer
c	Average outlet temperature	°F.	Calibrated Thermometer
d	Average temperature in system	°F.	(Item 5b + Item 5c) ÷ 2
e	Viscosity - cgs.	Centistokes	Fenske Viscosimeter
f	Viscosity - English units	Sq.Ft./Sec.	1.076 x 10^{-5} x Item 5e
g	Specific gravity 60°F./60°F.		Westphal balance
h	Specific gravity (based on 60°F. water)		Items 5c, 5d and curve
6	Manometer Constants		
a	Manometer fluid		
b	Density	g/cc	Westphal Balance
	at	°F.	Thermometer
c	Density	g/cc	Westphal Balance
	at	°F.	Thermometer
d	Connecting fluid		
e	Density at	g/cc	
f	Density at	g/cc	
g	Manometer tube area	sq.in.	
h	Reservoir area	sq.in.	
i	Manometer angle from horizontal	degrees	
j	Average manometer fluid temperature	°F.	Calibrated Thermometer
7	Oil Flow		
a	Average total flow for run	gallons	Calibrated Meter
b	Duration	minutes	Stop Watch
c	Average uncorrected rate	g.p.m.	Item 7a ÷ Item 7b
d	Meter constant		Item 7c and meter curve
e	Average corrected rate	g.p.m.	Item 7c x Item 7d
f	Average velocity	Ft./Sec.	0.4085 x Item 7e ÷ (Item 3b)^2
8	Reynolds Number		Item 3c x Item 7f ÷ Item 5f
9	Manometer Readings		
	Flow Readings - Entrance Section		
a	Leg A	Inches	
b	Leg B	Inches	
	No Flow Readings - Entrance Section		
c	Leg A	Inches	
d	Leg B	Inches	
	Flow Readings - Exit Section		
e	Leg C	Inches	
f	Leg D	Inches	
	No Flow Readings - Exit Section		
g	Leg C	Inches	
h	Leg D	Inches	
i	Pressure loss indication for entrance section	Inches	(Item 9a + Item 9b) - (Item 9c + Item 9d)
j	Pressure loss indication for exit section	Inches	(Item 9e + Item 9f) - (Item 9g + Item 9h)
10	Manometer Factors		
a	Difference in density of manometer and connecting fluids	g/cc	Items 6b, 6c, 5c, 6f, 6j, curve
b	Sine of manometer angle		Table
c	Density factor		Item 10a x Item 10b
d	Difference in specific gravity of connecting fluid and fuel oil 60°F./60°F.		Item 5g and Items 6e, 6f
e	Reservoir level change factor		Item 6g ÷ Item 6h
f	Correction to density factor		Item 10d x Item 10e
g	Corrected density factor		Item 10c + Item 10f
h	Multiplying factor for pressure		Item 10g ÷ 27.68
11	Equivalent Length of Fitting		
a	Pressure loss indication per foot of straight pipe (entrance section)		Item 9i ÷ 22.5
b	Pressure loss indication for straight pipe in section with fitting		Item 11a x Item 4c
a	Pressure loss indication for fitting		Item 9i - Item 11b
d	Equivalent length in feet		Item 11c ÷ Item 11a
e	Equivalent length in diameters		Item 11d ÷ Item 3c
12	Pressure Losses	Lbs./Sq.in.	
a	For straight pipe per foot (from entrance section)		Item 10h x Item 11a
b	For section with fitting		Item 10h x Item 9j
c	For straight pipe in section with fitting		Item 10h x Item 11b
d	For fitting		Item 12b - Item 12c
13	Friction Factor - Average		$\frac{\text{Item 3b x Item 12a}}{0.193 \text{ (Item 5b)(Item 7f)}^2}$

FIGURE 4. TEST CODE OF CALCULATIONS FOR INDIVIDUAL FITTINGS

then

$$E = \psi (Re, D) \qquad (3f)$$

Equation (3f) is the functional relationship desired and is composed of dimensionless groups only. It is valid for either laminar or turbulent flow since no restriction as to the character of the flow was imposed in the derivation.

Equivalent lengths measured to points inside the transition length are called pseudo-equivalent lengths, to points outside the transition length real equivalent lengths, (E_u).

It is now necessary to arrive at the explicit form of equation (3f). This requires experimental data since dimensional analysis cannot give the form of a relationship except in conjunction with an experiment or another assumption.

The test set-up previously described was checked to see that the friction factors for the two sections, AB (without piezometer rings between A and B) and CD (with piezometer rings between C and D) were the same. That this is the case is clearly shown by the plot of f against Re in Figure 5.

By means of the test set-up previously described, observations were made on E as a function of D keeping Re constant and then

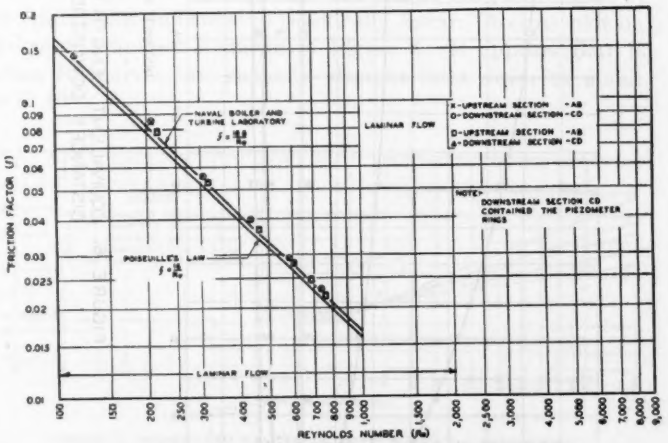

FIGURE 5. FRICTION FACTOR (f) VS. REYNOLDS NUMBER (Re) FOR STRAIGHT TUBING WITH AND WITHOUT PIEZOMETER RINGS

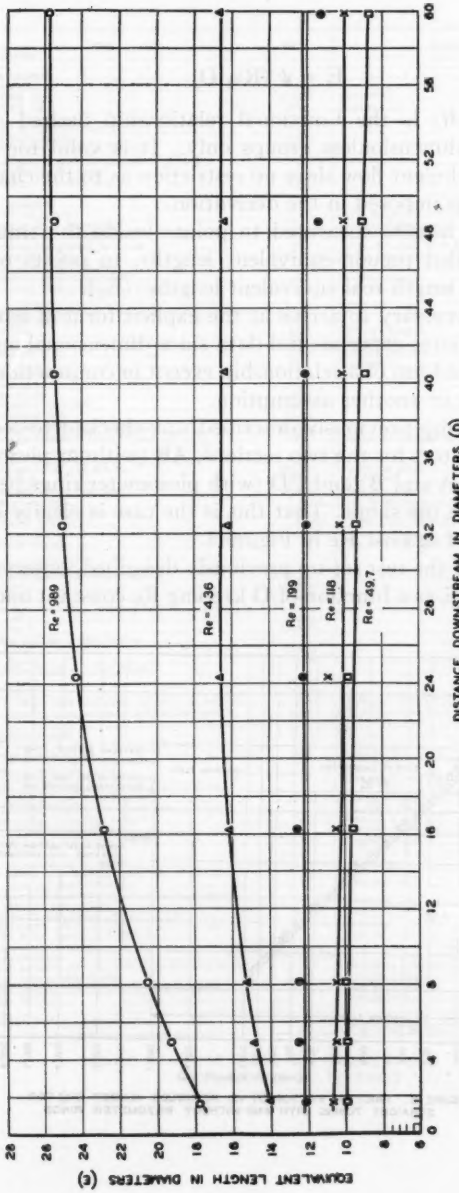

FIGURE 6. EQUIVALENT LENGTH FOR BEND R/d = 6 MEASURED AT VARIOUS DISTANCES DOWNSTREAM FOR CONSTANT REYNOLDS' NUMBER

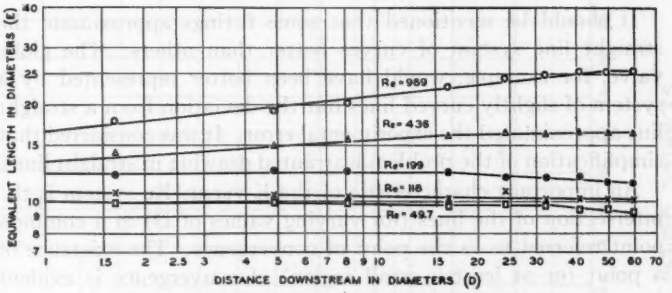

FIGURE 7. EQUIVALENT LENGTH FOR BEND R/d = 6 MEASURED AT VARIOUS DISTANCES DOWNSTREAM FOR CONSTANT REYNOLDS NUMBER

on E as a function of Re keeping D constant. The fittings tested are listed in the introduction. The test code is shown in Figure 4. Only bend R/d = 6 which has a fairly representative system of curves will be discussed in detail. Complete data, however, were taken on all the fittings, valves, and bends investigated.

In Figure 6 is the plot of equivalent length (E) versus distance downstream (D) for five Reynolds numbers ranging roughly from 50 to 1000. The same plot on log-log coordinate paper is presented in Figure 7. In the transition length, shown in Figure 7, the relationship indicated is practically linear. A cross plot on log-log coordinates is shown in Figure 8. It appears that E versus Re curves are virtually straight lines down to about Re = 100.

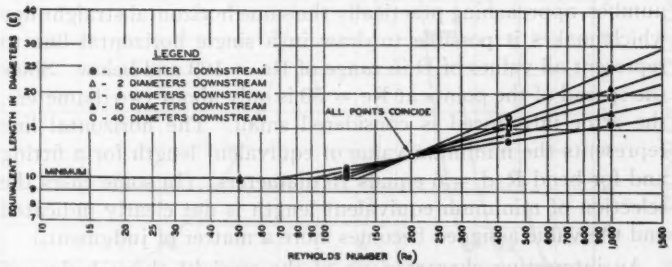

FIGURE 8 - EQUIVALENT LENGTH OF BEND R/d = 6 VS. REYNOLDS NUMBER FOR CONSTANT DISTANCES DOWNSTREAM

It should be mentioned that some fittings approximate the straight line system of curves better than others. The globe valve, for example, would have been better represented by a system of slightly curved lines but the deviation from a straight line approximated the experimental error. It was considered that simplification of the problem warranted drawing in straight lines.

An important characteristic of the E versus Re system is th intersection of the lines (for varying values of D) in a common point referred to as the point of convergence. The existence of a point (or at least a small region) of convergence is evident from the reversed positions of the E versus Re lines above and below the area of intersection. The equivalent length and Reynolds number corresponding to the convergent point are designated the convergent equivalent length (E_c) and the convergent Reynolds number (Re_c), respectively. For bend R/d = 6, E_c = 11.9 diameters and Re_c = 200.

The value of the ordinary equivalent length is called E_u. The distance from the fitting where the pseudo-equivalent length becomes equal to the real equivalent length is the transition length, D_u. For example, at Re = 989, bend R/d = 6 (Figure 6) has an E_u value of 25.7 diameters and a D_u value of approximately 41 diameters. The E_c' and Re_c quantities are characteristic of the fitting and the corresponding pipe or tubing. The E_u and D_u quantities depend on the type of fittings, the connecting pipe size, and the Reynolds number.

In the region of Re = 100, the slopes of the curves in the E versus Re system are slowly decreasing with decreasing Reynolds number, approaching practically the same horizontal straight line which makes it possible to draw in a single horizontal line to represent all values of D in range of Re = 100 and below. Since the spread of the points at Re = 50 is only about two diameters, the error introduced is considered small. The horizontal line represents the minimum value of equivalent length for a fitting and for bend R/d = 6 equals 10 diameters. (In some cases the selection of minimum equivalent length is not clearly indicated and the value assigned becomes more a matter of judgment.)

An interesting characteristic of the straight through flow of tee is the nature of its E vs. D system shown in Plate 9. Compared with the corresponding plot for the fittings (bend R/d = 6

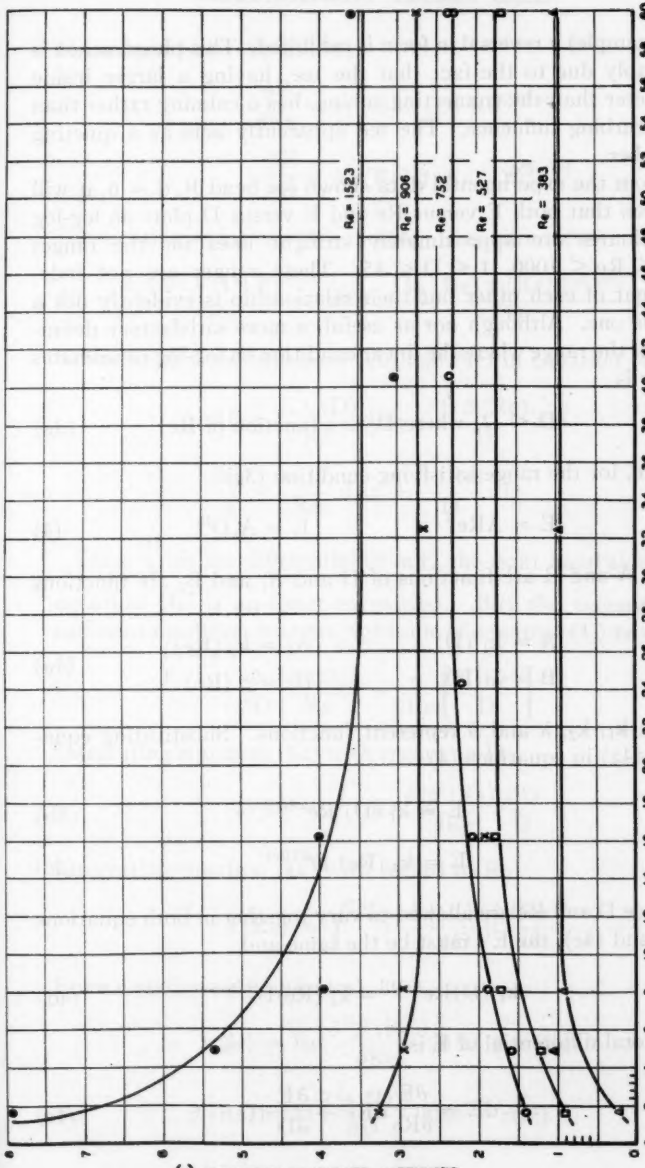

FIGURE 9. EQUIVALENT LENGTH FOR THROUGH FLOW OF TEE MEASURED AT VARIOUS DISTANCES DOWNSTREAM FOR CONSTANT REYNOLDS NUMBER.

for example) a reversal in form is exhibited. This phenomenon is probably due to the fact that the tee, having a larger inside diameter than the connecting tubing, has a calming rather than a disturbing influence. The tee apparently acts as a quieting chamber.

From the experimental data shown for bend R/d = 6, it will be seen that both E versus Re and E versus D plots on log-log coordinates are approximately straight lines for the ranges $100 < Re < 1000$, $1 < D < 45$. These ranges are not independent of each other but their relationship is evidently not a simple one. Although not as useful, a more satisfactory definition of the range where the linear condition on log-log coordinates exists is

$$D < D_u \text{ where } D_u \text{ is a function of Re} \qquad (3g)$$

Hence, for the range satisfying condition (3g)

$$E = ARe^B \qquad E = A_1 D^{B_1} \qquad (4)$$

Since A and B are functions of D and A_1 and B_1 are functions of Re

$$\left.\begin{aligned} A &= k_1 (D) & A_1 &= k_2 (Re) \\ B &= \lambda (D) & B_1 &= \beta (Re) \end{aligned}\right\} \qquad (4a)$$

where k_1, k_2, λ and β represent functions. Substituting equations (4a) in equations (4)

$$E = k_1 (D)\, Re^{\lambda (D)} \qquad (4b)$$

$$E = k_2 (Re)\, D^{\beta (Re)} \qquad (4c)$$

Since D and Re are allowed to vary together in both equations (4b) and (4c), the E's must be the same and

$$k_1 (D) Re^{\lambda (D)} = k_2 (Re) D^{\beta (Re)} \qquad (4d)$$

The total differential of E is

$$dE = \frac{\partial E}{\partial Re} dRe + \frac{\partial E}{\partial D} dD \qquad (4e)$$

From equations (4b) and (4c)

$$\frac{\partial E}{\partial Re} = k_1 (D)\, \lambda (D)\, Re^{\lambda(D)-1} \tag{4f}$$

$$\frac{\partial E}{\partial D} = k_2 (Re)\, \beta (Re)\, D^{\beta(Re)-1} \tag{4g}$$

Substituting equations (4f) and (4g) in equation (4e)

$$dE = \frac{k_1 (D)\, \lambda (D)\, Re^{\lambda(D)}}{Re} dRe + \frac{k_2 (Re)\, \beta (Re)\, D^{\beta (Re)}}{D} dD \tag{4h}$$

Substituting from equations (4b) and (4c) in equation (4h)

$$dE = \frac{E\, \lambda (D)}{Re} dRe + \frac{E\, \beta (Re)}{D} dD \tag{4i}$$

$$\frac{dE}{E} = \frac{\lambda (D)}{Re} dRe + \frac{\beta (Re)}{D} dD \tag{4j}$$

Since $\frac{dE}{E}$ is the differential of $ln\ E$, the right hand side of the equation (4j) is an exact expression. But the necessary and sufficient condition that the right side of equation (4j) be exact is

$$\frac{\partial}{\partial D}\left[\frac{\lambda (D)}{Re}\right] = \frac{\partial}{\partial Re}\left[\frac{\beta (Re)}{D}\right] \tag{4k}$$

Integrating equation (4k) with regard to D

$$\lambda (D) = \int^{D} Re \frac{\partial [\beta (Re)]}{\partial Re} \frac{dD}{D} \tag{4l}$$

Integrating equation (4k) with regard to Re

$$\beta (Re) = \int^{Re} D \frac{\partial [\lambda (D)]}{\partial D} \frac{dRe}{Re} \tag{4m}$$

From equations (4l) and (4m)

$$\lambda (D) = Re \frac{\partial [\beta (Re)]}{\partial Re} ln\ D + \tau (Re) \tag{4n}$$

$$\beta (Re) = D \frac{\partial [\lambda (D)]}{\partial D} lnRe + \tau(D) \tag{4p}$$

where τ represents a function. From equation (4k)

$$\mathrm{Re}\frac{\partial\,[\beta\,(\mathrm{Re})]}{\partial \mathrm{Re}} = \mathrm{D}\frac{\partial\,[\lambda\,(\mathrm{D})]}{\partial \mathrm{D}} \tag{4q}$$

Substituting equation (4q) in equation (4n)

$$\lambda\,(\mathrm{D}) = \mathrm{D}\frac{\partial\,[\lambda\,(\mathrm{D})]}{\partial \mathrm{D}}\,ln\,\mathrm{D} + \tau\,(\mathrm{Re}) \tag{4r}$$

Transposing and dividing by (D ln D) $[\tau\,(\mathrm{Re}) - \lambda\,(\mathrm{D})]$

$$\mathrm{O} = \frac{\mathrm{dD}}{\mathrm{D}\,ln\,\mathrm{D}} + \frac{\mathrm{d}\,[\lambda\,(\mathrm{D})]}{\tau\,(\mathrm{Re}) - \lambda\,(\mathrm{D})} \tag{4s}$$

Integration of equation (4s) gives

$$\lambda\,(\mathrm{D}) = c_1\,ln\,\mathrm{d} + \tau\,(\mathrm{Re}) \tag{4t}$$

Since equations (4n) and (4t) must be simultaneously true,

$$\mathrm{Re}\frac{\partial\,[\beta\,(\mathrm{Re})]}{\partial \mathrm{Re}} = c_1 \tag{4u}$$

Integrating equation (4u)

$$\beta\,(\mathrm{Re}) = c_1\,ln\,\mathrm{Re} + \mathrm{b}\ . \tag{4v}$$

Similarly it can be shown that

$$\lambda\,(\mathrm{D}) = c_1\,ln\,\mathrm{D} + \mathrm{h} \tag{4w}$$

Substituting from equations (4v) and (4w) in equation (4j) and transposing

$$\frac{(c_1\,ln\,\mathrm{d} + \mathrm{h})}{\mathrm{Re}}\mathrm{dRe} + \frac{(c_1\,ln\,\mathrm{Re} + \mathrm{b})}{\mathrm{D}}\mathrm{dD} - \frac{\mathrm{dE}}{\mathrm{E}} = \mathrm{O} \tag{5}$$

which is the differential equation relating E, D, and Re.

The necessary and sufficient condition that equation (5) be integrable is satisfied. Holding E constant, dE = O and

$$\frac{(c_1\,ln\,\mathrm{D} + \mathrm{h})}{\mathrm{Re}}\mathrm{dRe} + \frac{(c_1\,ln\,\mathrm{Re} + \mathrm{b})}{\mathrm{D}}\mathrm{dD} = \mathrm{O} \tag{5a}$$

Separating the variables

$$\frac{dRe}{Re\,(c_1\,ln\,Re + b)} + \frac{dD}{D\,(c_1\,ln\,D + h)} = O \qquad (5b)$$

The constant of integration will be a function of the third variable, E, since it is being held constant.

$$\int \frac{dRe}{Re\,(c_1\,ln\,Re + b)} + \int \frac{dD}{D\,(c_1\,ln\,D + h)} = \phi\,(E) \qquad (5c)$$

Integrating and transposing

$$\frac{1}{c_1}\,ln\,(c_1\,ln\,Re + b) + \frac{1}{c_1}\,ln\,(c_1\,ln\,D + h) - \phi\,(E) = O \qquad (5d)$$

Differentiating equation (5d)

$$\frac{dRe}{Re\,(c_1\,ln\,Re + b)} + \frac{dD}{D\,(c_1\,ln\,D + h)} - d\phi(E) = O \qquad (5e)$$

$$\frac{(c_1\,ln\,D + h)}{Re}\,dRe + \frac{(c_1\,ln\,Re + b)}{D}\,dD$$

$$- (c_1\,ln\,d + h)\,(c_1\,ln\,Re + b)\,d\,\phi\,(E) = O \qquad (5f)$$

Comparing equations (5) and (5f)

$$\frac{dE}{E} = (c_1\,ln\,D + h)\,(c_1\,ln\,Re + b)\,d\,\phi\,(E) \qquad (5g)$$

From equation (5d)

$$c_1\,\phi\,(E) = ln\,[(c_1\,ln\,D + h)\,(c_1\,ln\,Re + b)] \qquad (5h)$$

$$e^{c_1\,\phi\,(E)} = c_1\,ln\,d + h)\,(c_1\,ln\,Re + b) \qquad (5i)$$

Substituting equation (5i) in (5g)

$$\frac{dE}{E} = e^{c_1\,\phi\,(E)} d\,\phi\,(E) \qquad (5j)$$

Integrating equation (5j)

$$c_1\,ln\,a_1\,E = e^{c_1\,\phi\,(E)} \qquad (5k)$$

Taking the logarithm of both sides

$$ln\ ln\ (a_1\ E)^{c_1} = c_1\ \phi\ (E) \tag{5l}$$

Substituting for $\phi(E)$ in equation (5d)

$$\frac{1}{c_1}\ ln\ (c_1\ ln\ Re + b) + \frac{1}{c_1}\ ln\ (c_1\ ln\ D + h) = \frac{1}{c_1}\ ln\ ln(a_1E)^{c_1} \tag{5m}$$

Combining,

$$ln\ [(c_1\ ln\ Re + b)\ (c_1\ ln\ D + h)] = ln\ ln\ (a_1\ E)^{c_1} \tag{5n}$$

$$(c_1\ ln\ Re + b)\ (c_1\ ln\ D + h) = ln\ (a_1\ E)^{c_1} \tag{5p}$$

Multiplying through, substituting $c_1\ ln\ a_2 = hb$ and collecting

$$ln\ [Re^{c_1\ (c_1\ ln\ D\ +\ h)}\ D^{c_1\ b}\ a_2^{c_1}] = ln\ (a_1\ E)^{c_1} \tag{5q}$$

then

$$a_2\ D^b\ Re^{c_1\ ln\ D\ +\ h} = a_1\ E \tag{5r}$$

Dividing by a_1 and letting $a = \frac{a_2}{a_1}$

$$E = a\ D^b\ Re^{c_1\ ln\ D\ +\ h} \tag{5s}$$

which is the solution of the differential equation (5).

Converting from natural to Briggsian logarithms,

$$E = a\ D^b\ Re^{c\ log\ D\ +\ h} \tag{6}$$

which represents the relationship between E, D, and Re for a fitting and its corresponding pipe or tubing.

Equation (6) is called the fundamental equation and is the explicit form of the functional relation (3f) originally established by dimensional analysis. Since from the property of logarithms $Re^{c\ log\ D} = D^{c\ log\ Re}$, equation (6) may also be written

$$E = a\ Re^h\ D^{c\ log\ Re\ +\ b} \tag{6a}$$

To show the symmetry with regard to the variables Re and D, equation (6) may be written

$$E = \frac{a}{2} D^b \, Re^h \left[Re^{c \log D} + D^{c \log Re} \right]$$

Equations (6) and (6a) correspond to equations (4b) and (4c) and represent the surface in space shown in Figure 10. For simplicity, the three dimensions are indicated as logarithmic. The line AB is the convergent line and is perpendicular to the log E-log Re plane. The coordinates of this line for all values of D are log E_c and log Re_c.

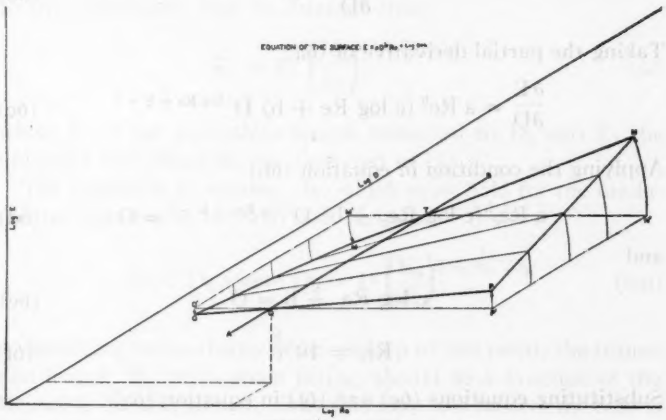

FIGURE 10. E-D-Re SURFACE

The coordinates of point Q on line AB are log E_c, log Re_c, and log D_1. The coordinates of point T are log E_c, log Re_c, and log D_2 where $D_2 > D_1$. Let OP, which is a straight line (since the E-Re lines on log-log coordinates are straight), be the generating element of the surface. Now let OP move so that Q slides toward T along the convergent line AB and at the same time let the slope of the generating element at any point be a linear function of the logarithmic distance the element has moved from Q. Such a motion will generate the surface OPNM and represents geometrically the fundamental equation.

Assuming that conventional pipe or tubing sizes are used, each fitting has its own fundamental equation and corresponding surface. It is apparent that the convergent point (E_c, Re_c) is characteristic of the fitting. Hence, it should be possible to determine the values of E_c and Re_c for a given fitting from the constants (a, b, c, h) of the system. The fundamental equation should also yield the relation of E_u to Re and a method for calculating D_u.

$$E = a\, Re^h\, D^{c \log Re + b} \tag{6a}$$

Since at the convergent point E is independent of D

$$\frac{\partial E}{\partial D} = O \tag{6b}$$

Taking the partial derivative of (6a)

$$\frac{\partial E}{\partial D} = a\, Re^h\, (c \log Re + b)\, D^{c \log Re + b - 1} \tag{6c}$$

Applying the condition in equation (6b)

$$a\, Re_c^h (c \log Re_c + b)\, D^{c \log Re_c + b - 1} = O \tag{6d}$$

and

$$c \log Re_c + b = O \tag{6e}$$

$$Re_c = 10^{-\frac{b}{c}} \tag{6f}$$

Substituting equations (6e) and (6f) in equation (6a)

$$E_c = a\, 10^{-\frac{hb}{c}} \tag{6g}$$

Equations (6f) and (6g) give the coordinates of the convergent point in terms of the constants of the system. The relation between E_c and Re_c is from equations (6f) and (6g),

$$E_c = a\, Re_c^h \tag{6h}$$

For a given Reynolds number it is now possible to calculate the transition length, D_u. Substituting in the fundamental equation (6a) $D = D_u$ and $E = E_u$.

$$E_u = a\, Re^h\, D_u^{c \log Re + b} \tag{6i}$$

Dividing equation (6i) by the fundamental equation (6a)

$$\frac{E_u}{E} = \left(\frac{D_u}{D}\right)^{c \log Re + b} \tag{6j}$$

$$D_u = D \left(\frac{E_u}{E}\right)^{\frac{1}{c \log Re + b}} \tag{6k}$$

Equation (6k) together with one of the E versus D plots for a fitting at a given Reynolds number gives the value of D_u for the prevailing flow conditions.

In general, if the equivalent length measured to one point in the transition length is known, the equivalent length measured to any other point may be obtained from

$$E_2 = E_1 \left(\frac{D_2}{D_1}\right)^{c \log Re + b} \tag{6l}$$

where E_1 is the equivalent length measured to D_1 and E_2 the unknown equivalent length measured to D_2.

The condition in relation (3g) which must hold for the fundamental equation to be valid now becomes

$$D < D_u \text{ where } D_u = D \left(\frac{E_u}{E}\right)^{\frac{1}{c \log Re + b}} \tag{6m}$$

According to the theory developed up to this point, the transition length, D_u, for a given fitting, should be a function of the Reynolds number alone. That this may not be strictly true is indicated by some experiments on bend R/d = 3.

The Reynolds number of 985 was obtained two different ways; first, with an oil temperature of 122°F. and second, with an oil temperature of 130°F. In Figure 11 it will be seen that the two different curves correspond to the two different oil temperatures. It should be noticed that the real equivalent length, E_u, is the same for both curves. The transition length, D_u, and the rate at which the pseudo-equivalent length becomes the real equivalent length, E_u, appear to differ for the two sets of flow conditions even though the Reynolds numbers are the same.

Whether these differences are rightfully chargeable to experimental error or whether this observation indicates that the forms

of the E versus D curves and the transition length, D_u, depend not only on the Reynolds number but also on the manner in which the Reynolds number is obtained remains for the present a matter for conjecture.

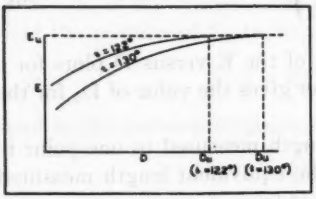

FIGURE 11

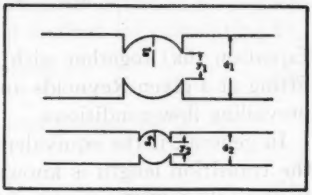

FIGURE 12

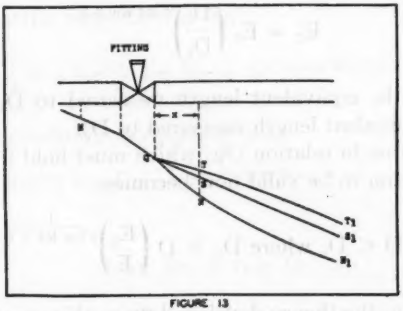

FIGURE 13

Were these effects great, it would follow that a, b, c, and h_a in the fundamental equation are not strictly constant for a particular fitting but only approximately so. They would depend on how the flow conditions for the Reynolds number are attained.

The problem of extending the graphical and mathematical solutions obtained for one fitting to a similar fitting of different size is important. It is evident that all dynamically similar systems will have the same solutions, i.e., the same value of equivalent length in diameters when measured to corresponding points.

Assume two systems with fittings of different sizes and corresponding tubing as shown in Figure 12. For the two systems to be dynamically similar,

$$E_1 = E_2;\ D_1 = D_2;\ Re_1 = Re_2 \tag{7}$$

where the subscript 1 refers to the larger fitting.

Substituting in the fundamental equation (6) for the two systems,

$$E = a_1 D^{b_1} Re^{c_1 \log D + h_1} \tag{7a}$$

$$E = a_2 D^{b_2} Re^{c_2 \log D + h_2} \tag{7b}$$

Dividing equation (7a) by (7b)

$$1 = \frac{a_1}{a_2} D^{b_1 - b_2} Re^{(c_1 - c_2) \log D + h_1 - h_2} \tag{7c}$$

Since D and Re are independent variables, equation (7c) can be true only if

$$a_1 = a_2,\ b_1 = b_2,\ c_1 = c_2,\ h_1 = h_2 \tag{7d}$$

The constants a, b, c, and h of a system are functions of the relative geometrical contours and roughness of the fitting and the tubing. The roughness factor is difficult to evaluate but it is likely that in the laminar region its effect is small compared with the obstruction offered by the physical contours of the fitting itself.

For the fittings to be geometrically similar it is necessary that the ratio of all corresponding dimensions be equal. Let y_1 and y_2 (Figure 12) be the bores of the fittings and z_1 and z_2 any other corresponding linear dimensions. Then

$$\frac{z_1}{y_1} = \frac{z_2}{y_2} \tag{7e}$$

For the systems to be similar, the ratios of the bores of the fittings to the inside diameter of the connecting tubing must also be equal. Let d_1 and d_2 be the diameter of the tubing, then

$$\frac{y_1}{d_1} = \frac{y_2}{d_2} \tag{7f}$$

The necessary and sufficient conditions that two systems be completely similar are given by equations (7), (7e) and (7f).

IV—Discussion of Flow Conditions in the Transition Length.

The E versus D lines for a fitting have a positive slope for Reynolds numbers above Re_c, substantially a zero slope for Reynolds numbers equal to Re_c and a negative slope for Reynolds numbers under Re_c.

When this difference in the sign of the slope was first observed, it was difficult to visualize the existence of a critical value, Re_c, because the investigated flow is laminar above and below the convergent Reynolds number. Since Poiseuille's Law fits the laminar region and since the flow in the straight pipe is the same, the explanation must come from the manner in which the fluid is affected in passing through the fitting.

Figure 13 shows the pressure gradients of a system at $Re > Re_c$ ($MQNN_1$), at $Re = Re_c$ ($MQSS_1$) and at $Re < Re_c$ ($MQTT_1$) where T, S, and N are points in the transition length.

Let $\frac{dp_M}{dx}$ be the pressure gradient at point M and characteristic of the particular flow conditions. From the slope of the E vs. D curves in the transition range, the following relationships for points T, S, and N must be true.

$$\text{when } Re < Rc_c, \frac{dp_T}{dx} < \frac{dp_M}{dx} \tag{8a}$$

$$\text{when } Re = Re_c, \frac{dp_S}{dx} = \frac{dp_M}{dx} \tag{8b}$$

$$\text{when } Re > Re_c, \frac{dp_N}{dx} > \frac{dp_M}{dx} \tag{8c}$$

The experimental fact in relation (8a) indicates an increase in static pressure at point T in the transition length. For a fixed flow rate, the only possible way to increase the static pressure is by conversion at the expense of velocity pressure. From Bernoulli's Theorem, the transformation of velocity pressure to static pressure occurs when the cross-sectional area of the moving fluid is increasing. With this fact in mind, it is possible to picture the conditions of flow in the transition length.

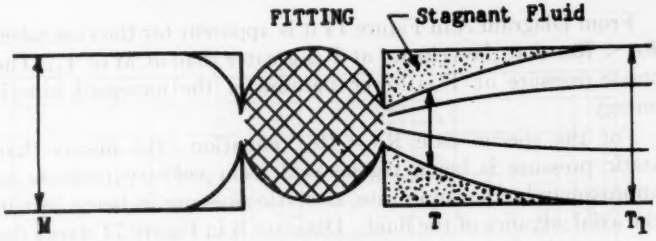

DIAGRAM A

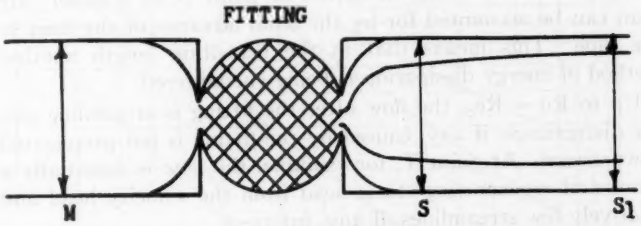

DIAGRAM B

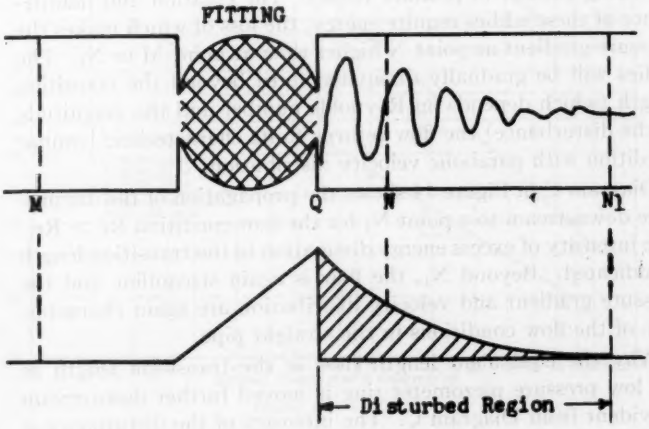

DIAGRAM C

FIGURE 14

From Diagram A in Figure 14 it is apparent for the case when $Re < Re_c$ that the velocity at T is greater than at M or T_1. The static pressure at T is lower because of the increased kinetic energy.

For the special case $Re = Re_c$, equation (8b) means that static pressure is being regenerated from velocity pressure at approximately the same rate as static pressure is being lost in the axial advance of the fluid. Diagram B in Figure 14 shows the cross-sectional area at S, any point downstream from the fitting, to be the same as at M and S_1.

For the last case, $Re > Re_c$, equation (8c) indicates that static pressure is being dissipated at point N at a faster rate than can be accounted for by the axial advance of the fluid in the pipe. This means that in the transition length another method of energy dissipation is being encountered.

Up to $Re = Re_c$, the flow after the fitting is streamline and the disturbance, if any, caused by the fitting is not propagated downstream. At point T, for example, the flow is essentially a process of regenerating static head from the velocity head and relatively few streamlines, if any, intersect.

For the case where $Re > Re_c$, the disturbance caused by the fitting is propagated downstream indicating that the streamlines are being broken to produce eddies. The creation and maintenance of these eddies require energy, the loss of which makes the pressure gradient at point N higher than at point M or N_1. The eddies will be gradually dissipated until beyond the transition length (which depends on Reynolds number and the magnitude of the disturbance) the flow returns to its characteristic laminar condition with parabolic velocity distribution.

Diagram C in Figure 14 shows the propagation of the disturbance downstream to a point N_1 for the flow condition $Re > Re_c$. The intensity of excess energy dissipation in the transition length is indicated. Beyond N_1, the flow is again streamline and the pressure gradient and velocity distribution are again characteristic of the flow conditions in the straight pipe.

Why the equivalent length rises in the transition length as the low pressure piezometer ring is moved further downstream is evident from Diagram C. The intensity of the disturbance is falling off but more of the energy lost is being measured until at

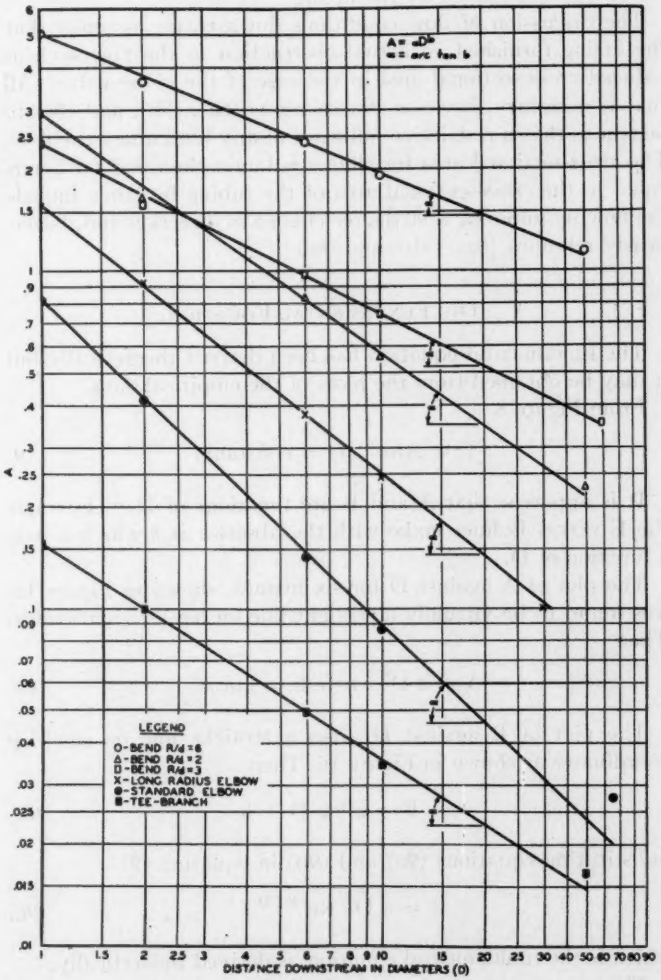

FIGURE 15. A VS. D

point N_1 and beyond the pressure measurement will include all the energy loss caused by the fitting.

The discussion of flow conditions downstream assumed that the fitting furnished an actual obstruction to the flow such as reduced cross-sectional area in the case of the globe valve. All that is necessary, however, for relations (8a), (8b), and (8c) to be true is that a restrictive influence of any kind affect the flow. The cross-sectional area for elbows is larger than and for bends equal to the cross-sectional area of the tubing but they impede the flow by imposing a 90 degree change in direction and, hence, satisfy relations (8a), (8b), and (8c).

V—The Fundamental Equation.

The fundamental equation has been derived theoretically but it may be obtained from the plots of the empirical data.

From Figure 8

$$E = ARe^B \text{ (D = constant)} \tag{9}$$

It is apparent that A and B are functions of D. The angle the E versus D lines make with the abscissa is θ which is also a function of D.

The plot of A against D for six fittings, shown in Figure 15, was found to be virtually a straight line on log-log coordinates. Then

$$A = a\,D^b \text{ where } b = \tan \alpha \tag{9a}$$

The plot of B against D gives a straight line on semi-log coordinates as shown in Figure 16. Then

$$B = c \log D + h \tag{9b}$$

Substituting equations (9a) and (9b) in equation (9)

$$E = a\,D^b\,Re^{c \log D + h} \tag{9c}$$

which is the fumdamental equation as derived theoretically.

The constants a and b can be evaluated from the A versus D plot, the constants c and h from the B versus D plot. Calculations for the six fittings and their fundamental equations are

shown in Figure 17. Comparisons of the calculated and observed values of E_c and Re_c for the six fittings are also shown in Figure 17. In view of the large degree of uncertainty in the constants a, b, c, and h, the calculated and observed values of E_c and Re_c agree fairly well.

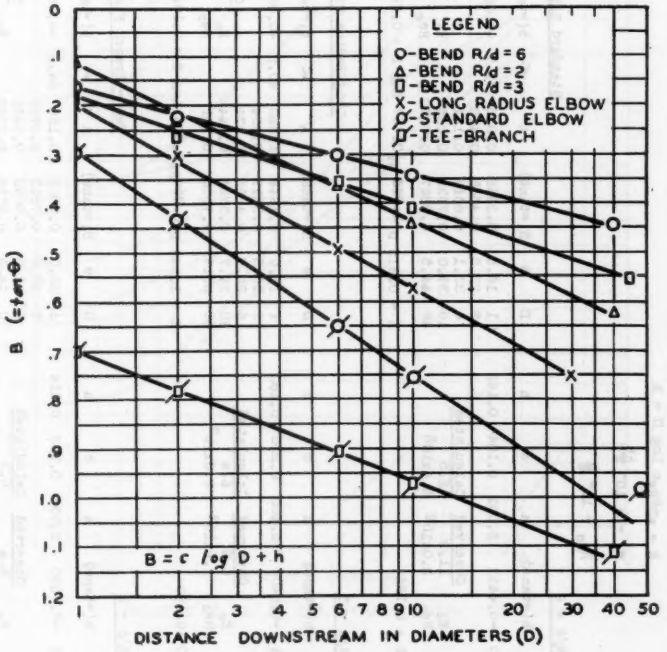

FIGURE 16. B VS. D

The globe and angle valves do not show enough variation of E with D to derive their fundamental equations. The equivalent lengths of the gate valve and through-flow of the tee are too small and inconsistent to permit the determination of the constants of the fundamental equation.

CALCULATION OF CONSTANTS

Equations Involved:

$E = ARe^B$ in which $A = aD^b$ and $B = c \text{ Log } D + h$

$E = aD^bRe^c \text{ Log } D + h$

$K_c = a\ 10^{-\frac{bh}{c}}$

$Re_c = 10^{-\frac{h}{c}}$

Bend R/d = 6

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	9.2	0.1620	5.051	24.0	-0.4452	5.051	0.184	0.162
2	12.7	0.2254	3.214					
6	16.7	0.3000	2.435					
10	19.1	0.3463	1.906					
40	23.9	0.4431	1.143					

	Observed	Calculated
K_c	11.9	12.5
Re_c	2.0×10^2	2.6×10^2

$E = 5.051\ D^{-0.445}\ Re^{0.184} \text{ Log } D + 0.162$

Bend R/d = 3

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	10.2	0.1799	2.400	28.4	-0.5407	2.400	0.230	0.180
2	14.8	0.2642	1.565					
6	19.7	0.3581	0.971					
10	22.3	0.4101	0.746					
45	29.1	0.5566	0.355					

	Observed	Calculated
K_c	6.9	6.4
Re_c	1.6×10^2	2.3×10^2

$E = 2.400\ D^{-0.541}\ Re^{0.230} \text{ Log } D + 0.180$

Bend R/d = 2

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	6.5	0.1139	2.938	37.2	-0.7590	2.938	0.333	0.114
2	12.9	0.2290	1.653					
6	20.2	0.3679	0.826					
10	24.1	0.4473	0.555					
40	32.0	0.6249	0.229					

	Observed	Calculated
K_c	5.4	5.3
Re_c	1.5×10^2	1.9×10^2

$E = 2.938\ D^{-0.759}\ Re^{0.333} \text{ Log } D + 0.114$

Standard Ell

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	16.5	0.2962	0.815	45.9	-1.0319	0.815	0.457	0.296
2	23.5	0.4348	0.412					
6	33.1	0.6519	0.141					
10	37.0	0.7536	0.0857					
48	44.5	0.9827	0.0270					

	Observed	Calculated
K_c	3.5	3.8
Re_c	1.4×10^2	1.8×10^2

$E = 0.815\ D^{-1.038}\ Re^{0.457} \text{ Log } D + 0.296$

Long Radius Ell

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	11.0	0.1944	1.568	40.5	-0.8541	1.568	0.376	0.194
2	16.8	0.3019	0.924					
6	25.9	0.4856	0.374					
10	29.7	0.5704	0.244					
30	36.9	0.7508	0.101					

	Observed	Calculated
K_c	4.1	4.3
Re_c	1.4×10^2	1.9×10^2

$E = 1.568\ D^{-0.854}\ Re^{0.376} \text{ Log } D + 0.194$

Tee - Branch Flow

D	θ	B(=tanθ)	A	∝	b(=tan∝)	a	c	h
1	35.0	0.7002	0.155	34.6	-0.6899	0.155	0.276	0.700
2	38.0	0.7813	0.0999					
6	42.3	0.9099	0.0491					
10	44.3	0.9759	0.0348					
40	48.1	1.1145	0.0165					

	Observed	Calculated
K_c	6.8	8.5
Re_c	2.2×10^2	3.2×10^2

$E = 0.155\ D^{-0.690}\ Re^{0.276} \text{ Log } D + 0.700$

FIGURE 17. FUNDAMENTAL EQUATION - EMPIRICAL DETERMINATION.

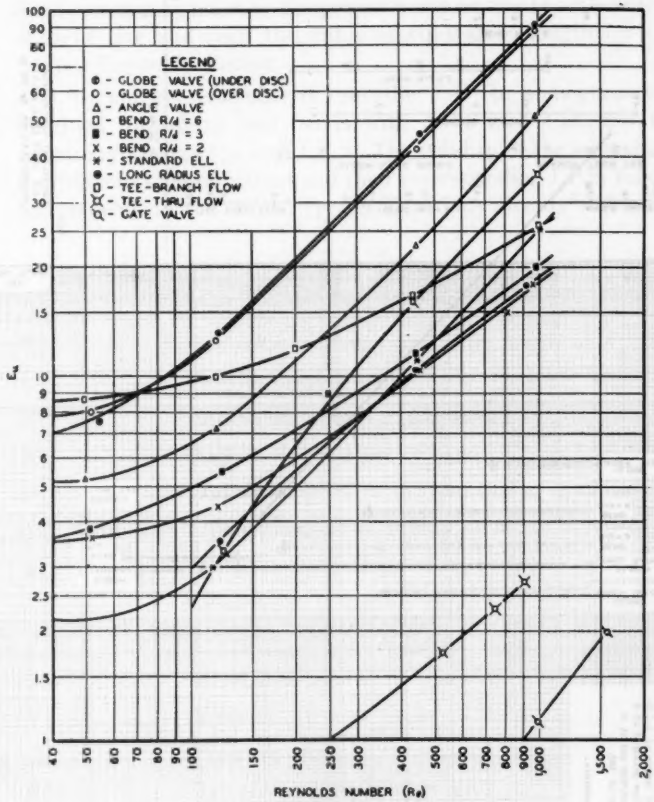
LEGEND
⊕ - GLOBE VALVE (UNDER DISC)
○ - GLOBE VALVE (OVER DISC)
△ - ANGLE VALVE
□ - BEND R/d = 6
■ - BEND R/d = 3
X - BEND R/d = 2
✱ - STANDARD ELL
● - LONG RADIUS ELL
▯ - TEE-BRANCH FLOW
⊠ - TEE-THRU FLOW
◇ - GATE VALVE
E_u
REYNOLDS NUMBER (R_e)

FIGURE 18. E_u VS. R_e FOR FITTINGS TESTED

VI—Equivalent Length Nomogram.

The plots of the real equivalent length (E_u), against Reynolds number for all the fittings tested are shown in Figure 18.

A nomogram of real and pseudo-equivalent lengths, prepared from the original data of the fittings, is presented in Figure 19. The minimum values of equivalent length are also indicated. To use the E-D-Re nomogram for a particular fitting, connect the

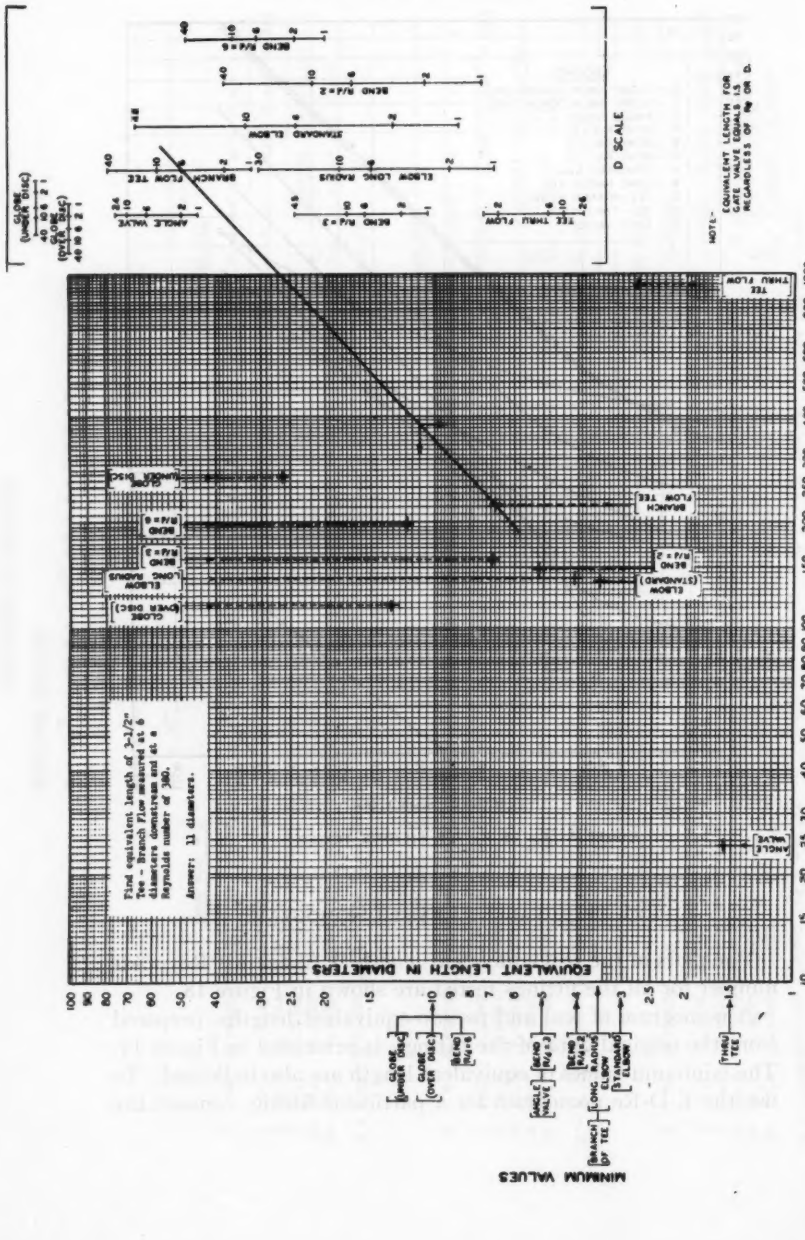
D SCALE
NOTE:-
EQUIVALENT LENGTH FOR GATE VALVE EQUALS 13 REGARDLESS OF Re OR D.
EQUIVALENT LENGTH IN DIAMETERS
REYNOLDS NUMBER Re = VD/ν
MINIMUM VALUES
Find equivalent length of 3-1/2" Tee - Branch Flow measured at 6 diameters downstream and at a Reynolds number of 360.
Answer: 11 diameters.
GLOBE (UNDER DISC)
GLOBE (OVER DISC)
BEND R/d = 6
BEND R/d = 3
BEND R/d = 2
ANGLE VALVE
BRANCH FLOW TEE
LONG RADIUS ELBOW
STANDARD ELBOW
THRU TEE
GATE VALVE

FIGURE 19. E-D-Re NOMOGRAM

fixed convergent point to the proper point on the D scale with a straight line and read the value of equivalent length for the desired Reynolds number.

It should be remembered that the E-D-Re nomogram was obtained for tubing and bends with 3.705 inch I.D. and the remaining fittings 3½ inch I.P.S. The validity of the nomogram for different sizes of fittings and their corresponding I.P.S. tubing will depend on the similarity considerations already discussed.

GLOSSARY OF SYMBOLS AND TERMS.

Re	Reynolds number.
f	Friction factor.
E	Equivalent length in diameters.
R	Radius of curvature of bends.
D	Distance downstream from fitting in diameters.
d	Inside diameter of tubing.
x	Distance downstream from fitting.
ρ	Density.
μ	Viscosity.
μ/ρ	Kinematic viscosity.
V	Velocity.
Δp	Pressure drop.
R/d	Relative radius, bend ratio.
E_c	Equivalent length at convergent point.
Re_c	Reynolds number at convergent point.
Δp_o	Pressure drop of a fitting.
$_M[\Delta p]_N$	Pressure drop from a point M to a point N.
Δp_p	Pressure drop due to the straight pipe or tubing in a system.
Δp_c	Pressure drop per diameter of pipe or tubing.
E_u	Equivalent length when E is independent of D.

D_u	Distance downstream in diameters where E becomes independent of D.
X	Dimensionless group $\frac{V\rho x}{\mu}$.
$\psi, \psi_1, \psi_2, \psi_3, \lambda, \beta, k_1, k_2, \gamma, \phi$.	Functions.
a_1, a_2, k, c_1	Constants.
θ	Angle E vs. Re lines make with abscissa on log-log paper.
A	Coefficient in equation representing E vs. Re lines ($E = A\,Re^B$).
a	Coefficient in fundamental equation.
B	Tangent θ.
α	Angle of A vs. D lines on log-log paper.
b	Tangent α, constant in fundamental equation.
c	Slope of B vs. D lines, constant in fundamental equation.
$\frac{dp}{dx}$	Pressure gradient.
y_1, y_2	Characteristic linear dimension of a fitting (example, bore of globe valve).
z_1, z_2	Any linear dimension of fitting.
d_1, d_2	Inside diameter of pipe or tubing.
Δp_u	Pressure drop corresponding to E_u.
Convergent point	The point on the E-Re plane where all E vs. Re lines converge. At this point, E is practically independent of D.
Diameters downstream	The distance downstream in diameters from a fitting to the downstream piezometer ring or the next fitting.
Fitting	General term for valve, fitting or bend; also called a unit.
Fundamental equation	$E = a\,D^b\,Re^{c \log D + h}$.

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GAS TURBINES AND SUPERCHARGERS.

Several illustrations and all references of this paper have been eliminated when reprinting it from the December, 1943, issue of General Electric Review. Students of the gas turbine are referred to that magazine for an excellent list of early references on the subject. This article was presented as a paper at the recent annual meeting of the American Society of Mechanical Engineers, in New York City, by Sanford A. Moss, long-time consulting engineer, Supercharger Engineering Division, General Electric Company.

Production of power by direct action of products of combustion on a turbine wheel has been an engineering dream for a century and a half. A number of approaches to the complete scheme now are in successful commercial operation, and it seems possible that these are paving the way for the long-sought accomplishment of the primary plan of the gas turbine as an efficient prime mover. So it would now seem opportune to give some history and reminiscence (without any thermodynamics), list the partial accomplishments now in commercial use, and speculate on the future. The author can do this from personal association with many of the items ever since he became, in 1895, one of the many who have had dreams in the matter.

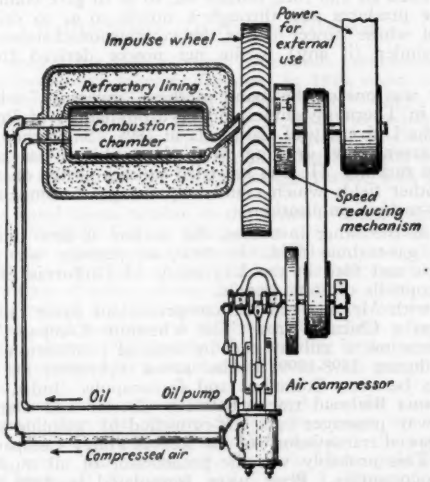

FIGURE 1.—DIAGRAMMATIC PLAN OF FUNDAMENTAL GAS TURBINE.

This association with gas-turbine developments has occurred during an engineering career which began in San Francisco in 1889, with a machinist apprenticeship, following with work as draftsman in the shop of Edward A. Rix (an early master in compressed-air engineering); and continuing as a draftsman in various gas-engine shops. This was before the days of electric transmission, and there were several plans of compressed-air power and transmission in successful operation. One was the use by Mr. Rix of the early impulse water-wheel of Pelton to drive an air compressor,

with a transmission pipe line some miles long, and then a reciprocating hoisting engine using the transmitted air after it had been reheated by coal or coke so as to increase its volume. So this really was a combustion heat-engine cycle. Its thermodynamics, which is that of the "Brayton constant-pressure cycle," is an early item of the present subject. Mr. Rix and his associates also had a street railway in New York City operating on compressed air stored in steel cylinders. But the author remembers how these and similar schemes soon were replaced by the early electric transmissions.

Gas-turbine Proposals.

Beginning about 1890, both the steam turbine and the internal-combustion reciprocating engine began the competition which finally resulted in the elimination of the reciprocating steam engine as a prime mover, except for the steam locomotive, and some ship installations.

Ever since the beginning of this period, it keeps occurring independently to students of thermodynamics that the steam turbine and internal-combustion engine might be combined, and so they "invent" the gas turbine. The usual form of this, shown in Figure 1, has a chamber into which is forced compressed air and fuel, usually oil, so as to give combustion under pressure. The products pass through a nozzle so as to drive a turbine wheel, part of whose power drives the combustion-chamber compressor, and the remainder (if any) is the net power derived from the fuel burned.

The author was one of the many of these "inventors," while a student in the classes in Thermodynamics and Hydrodynamics of Prof. Frederick G. Hesse at the University of California in 1895. Since then, the author's engineering career, now drawing to a close, always has been directed toward the gas turbine. He also has tried to keep posted on the numerous advances in other fields which could help the gas turbine, as well as on the many gas-turbine publications.

Like most of the other inventors, the author at first thought he was alone in the gas-turbine field. In 1900, an account was written as a Master's Thesis and filed in the University of California Library, which gives early proposals of some details.

Associated with Mr. Rix and the compressed-air street railway in New York City was a Chicago firm, "The Vimotum Company," engaged in promoting operation of railway cars by internal combustion. The author participated during 1898-1899 in the actual operation on the "Big 4" railway tracks between Columbus and Indianapolis (Ind.), and later on the Pennsylvania Railroad tracks between Chester and Darby (Pa.), of full-sized railway passenger cars, self-propelled by gasoline engines, using the "variable-speed transmission" of the Reeves Pulley Company of Columbus (Ind.). This probably was the predecessor of all modern internal-combustion locomotives. Plans were formulated in 1898 for replacing the gasoline engine by a gas turbine as was set forth in the Master's Thesis. This is the earliest proposal that the author knows of for a gas-turbine locomotive. But the proposition never came to a head, which was just as well, because then it never could have been made operative.

During the planning for this proposed gas-turbine locomotive, patent searches were made, and unexpectedly there was found a British patent of November 30, 1791, to John Barber, giving probably for the first time, the fundamental gas-turbine plan. There was a combustion chamber into which fuel and air were introduced, with a nozzle from which the products of combustion issued onto a turbine wheel. This supplied power to drive

the compressor, as well as external power. Barber's apparatus probably never was operated.

An early diagram of Barber's cycle is given in Figure 1, from the Doctor's Thesis. Many gas turbines proposed since differ in plan only by the addition of a heat interchanger for transferring the heat of the exhaust gases to the compressed air proceeding to the combustion chamber. An early mention of this is in the Theses, and an experimental model was made in 1905.

Davey lists some partial approaches to the gas turbine and gives a list of British patents beginning in 1856. The earliest complete United States gas-turbine patent of which the author has personal knowledge is to Charles G. Curtis, June 24, 1895. Mr. Curtis had extensive connections, with regard to steam turbines, with the General Electric Company. The author has some knowledge of these, but none of any actual work on gas turbines by Mr. Curtis.

At this same time is a United States patent to Leon Le Pontois, filed December 5, 1895, on an explosive-combustion chamber for a gas turbine. This was the subject of a Doctor's Thesis at Cornell University by Walter O. Amsler in 1897. Actual tests were made of Le Pontois' explosive apparatus, as well as a theoretical study, with unfavorable conclusions.

United States Gas-turbine Research.

The author always has had in mind continuous combustion, and after the gas-turbine-locomotive scheme, proposed in 1898, came to naught, he started gas-turbine research in 1901, in the Sibley College Laboratory, Cornell University. Laboratory time for a year was required to get a continuous-combustion chamber in stable operation. It frequently went out and then the oil on the red-hot firebrick lining filled the neighborhood with dense black smoke, so the Sibley College people well knew of the gas-turbine research.

The first De Laval steam turbine in the United States, which had been exhibited at the World's Fair in Chicago in 1893, was used for the De Laval nozzle-patent demonstration mentioned later, and then went to Cornell. So the bucket wheel of this turbine was borrowed and operated with the combustion chamber, about the end of 1902. This probably was the first turbine wheel actually operated by products of combustion, in the United States, and possibly was the first such turbine wheel ever operated. The entire power was absorbed by a Prony brake, and the air for compression was furnished by a steam compressor, with computation of the power required.

But, as with many other experimental gas turbines, the power for compression was more than the turbine power. So except for the historical fact that the combustion chamber actually operated the turbine wheel, the experiment was a flat failure. A Doctor's Thesis was prepared and presented to Charles P. Steinmetz and Ernest J. Berg of the General Electric Company. The author previously had been a draftsman on steam-turbine work with General Electric. In June, 1903, the connection was resumed and has continued ever since, with work on various items more or less connected with gas turbines.

A number of General Electric models of gas turbines successively were gotten into operation, first at Schenectady, and, beginning in 1904, at Lynn, with consultation with Prof. Elihu Thomson and Richard H. Rice. There were combustion chamber, heat interchanger, nozzle, and single-stage impulse turbine wheel; but with power for compression from an independent air compressor and allowed for by computations. Some U. S. patents

were obtained during this research. The results of this research will be given later in this article. The patent drawings show fairly well the actual apparatus used.

Foreign Gas-turbine Research.

A gas-turbine patent was applied for by Charles Lemale about 1901, and he began association with Rene Armengaud in 1903, in actual operation in France of a continuous-combustion gas turbine. This probably was the earliest elaborate gas-turbine experimental work. From then on, there have been many proposals and tests of various sorts of gas turbines. Some of these operate with a hydraulic column; operate below atmospheric pressure; have cooling by water injection; have hollow turbine buckets with cooling; have cooling by air excess; use pulverized coal; use reheating combustion; or have cooled compressors. But most of these have not led to any conclusive results. The cycle now receiving the most attention is that of Figure 1, with the addition of the heat interchanger of the previously mentioned Theses.

Nozzles for Steam and Gas Turbines.

Beginning about 1885, there were many arguments concerning the power that could be obtained by expansion in a nozzle of a jet directed on a turbine wheel. Napier had shown experimentally that, when steam was discharged from an opening in a vessel into a region of lower pressure, the flow increased as the pressure in the outer region decreased, until the outer pressure became about one-half of the pressure in the vessel. For lower values of the outer pressure, there was no increase in flow. Therefore, it was argued that full theoretical velocity of a steam-nozzle jet could not be obtained when the pressure ratio was less than 0.5. It seemed reasonable to assume that the theoretical power from a steam-turbine wheel was the same as from a reciprocating steam engine, for pressure ratios between 1.0 and 0.5. But Napier's law meant that for pressure ratios less than 0.5 a steam turbine would not theoretically give the same power as a steam engine.

Carl Gustaf Patrick De Laval was building single-stage steam turbines in Sweden with steam pressures of 115 psia or so, exhausting into atmospheric pressure of about 15 psi. He said he was avoiding the disability of Napier's law by using a nozzle first convergent and then divergent, popularly called an "expanding nozzle," because of the final divergent portion. De Laval tried to get a United States patent on this shape, and the official file shows that Napier's law was cited against him, with the support of scientific experts, including Gustave Zeuner, the famous German engineer. Finally, De Laval with his 1893 Chicago World's Fair turbine gave an actual demonstration that he was right.

The theoretical reasons for a nozzle first convergent and then divergent already had been given by Osborne Reynolds, with the first diagram of such a nozzle. But this was buried in a philosophical magazine, and does not seem to have been appreciated by engineers. At a much later date it began to be understood that the theoretical available energy for a steam turbine with proper nozzles is the same as for the corresponding reciprocating engine. But there was a period within the memory of the author when engineering ignorance in this matter held back steam-turbine development.

Next it was argued that, while proper power could be developed by use of a steam nozzle, it was because of proximity of the fluid to condensation.

But, so it was said, circumstances differed with gases far removed from condensation, such as products of combustion, so that a gas-turbine nozzle never could be efficient. Experimental measurements were published of temperatures measured along a nozzle producing a jet of a nearly perfect gas, showing that the temperature did not fall with lowering of pressure, as was predicted by the formulas for isentropic expansion. It was pointed out that, with an expanding gas, mechanical energy only could be developed by conversion of internal energy, as manifested by temperature drop. Thus, it was argued that proper conversion of pressure drop into velocity could not be accomplished by a gas-turbine nozzle.

To give further support to this contention, a test was published in which a De Laval single-stage turbine was operated at full speed with compressed air. To avoid complication, no power was abstracted from the turbine shaft. There was measurement of temperature in the pipe preceding the nozzle, and in the exhaust pipe. The pressures in the inlet pipe varied in different tests from about 12 to 123 psi, with atmospheric pressure in the exhaust pipe. But in spite of these pressure drops, which corresponded to appreciable drops of isentropic temperature, the temperatures in the exhaust pipe always were nearly the same as in the inlet pipe. Since compressed air within the pressure limits used is nearly a perfect gas, the equality of temperature showed that, in spite of passage through the nozzle, no internal energy had been abstracted corresponding to an isentropic drop of temperature. Thus, a nearly perfect gas such as air or products of combustion would not liberate its energy by isentropic drop in a turbine nozzle, and so a gas turbine could not be efficient.

This experiment was published in all seriousness as a demonstration of the point made. But if a student of engineering has not by now detected the fallacy, he has fallen into a trap which often before has been set by the author for the unwary thermodynamist. As was mentioned with planned casualness, no energy was extracted from the turbine shaft, so that all of the kinetic energy of the jet, which no doubt was nearly the theoretical isentropic value, was converted into heat by turbine-bucket and rotation-loss friction. With the nearly perfect gas used, this restored the original temperature, as could be expected.

The temperature-measurement experiment was equally fallacious. The jet flowing along the nozzle no doubt had the full velocity and low temperature at each point corresponding to the isentropic pressure drop. But the temperature-measuring apparatus destroyed this velocity, converting it into heat, and so, with a perfect gas, restored the initial temperature.

The theory of this, possibly for the first time, was published in 1916. This paper also describes an experiment made by the author about 1905. Steam, initially slightly wet, and at an appreciable pressure, was discharged into the atmosphere through a properly shaped nozzle. As is well known, the initially wet steam became wetter as it expanded to atmospheric pressure, and so the jet leaving the nozzle had the "static" temperature of wet steam at atmospheric pressure, which is 212 F. But a thermometer inserted in the nozzle jet showed a much higher temperature, the result of conversion of some of the jet velocity into heat. Complete conversion would give a temperature called "impact temperature" in the 1916 reference, but now beginning to be called "total temperature." The situation nowadays is fairly well understood, and it seems to be remembered that it was explained in the 1916 publication of the 1905 experiment.

But, although the fallacies of the two publications mentioned are very evident now, and cannot be cited against the ultimate success of the gas turbine, and were partially refuted at the time, they were disturbing factors during the General Electric gas-turbine research and had to be argued against.

Compressors for Gas Turbines.

A part of a gas-turbine plant is an air compressor to force air into the combustion chamber, which must be driven by part of the power from the turbine wheel. In the fall of 1903, the General Electric gas-turbine research began to include the development of a "centrigufal compressor," with rotative speed suitable for direct connection to the gas-turbine wheel. High-speed wheels previously had been proposed for air compression, with some sort of "reversed nozzle," since called a "diffuser," for converting into pressure the absolute velocity of air leaving the wheel periphery. Such diffusers had begun to be used for incompressible fluids. They probably first had been used in 1895 in a centrifugal pump of the famous English scientist Osborne Reynolds.

But De Laval's plan of a turbine nozzle, first convergent and then divergent, for a compressible fluid, now was firmly fixed. With this was the idea that a "reversed nozzle" for converting velocity into pressure with a compressible fluid, such as air, always should be the reverse of the De Laval expanding nozzle. Therefore, the foregoing compressor references show such passages beyond the high-speed wheels, with the minimum section or throat of De Laval; and speak of "throat conversions of velocity into pressure." The famous French engineer, August Rateau, was working on high-speed centrifugal air compressors about this time, but the author never has been able to find out if he then used diffusers, and if so what his early ideas were about their shape. Rateau supplied centrifugal compressors for the Armengand-Lamale gas-turbine work.

In a patent applied for in 1904, it was set forth correctly, probably for the first time, that with velocities below the velocity of sound, in the case of a diffuser for a compressible fluid, the velocity changes are the major items, so that passage areas change inversely as velocities, just as with an incompressible fluid. But with velocities greater than the velocity of sound, the change of volume with a compressible fluid, due to change of pressure, is the major item, so that increase of velocity in a nozzle requires increase of area, and decrease of velocity in a diffuser requires decrease of area. So with a centrifugal air compressor with impeller exit velocity less than the velocity of sound, the diffuser must be divergent, just as with a centrifugal pump. But for velocities greater than the velocity of sound, the theoretical diffuser first is convergent, has a throat, and then is divergent. All of this now is elementary knowledge but in 1904 it was not evident.

The velocities of the General Electric centrifugal compressor for the proposed gas turbine always were such as to require theoretically a divergent diffuser, and development work on this basis was begun in 1904, under the supervision of Richard H. Rice, with the collaboration of the author and other General Electric engineers. When the primary gas-turbine research ceased in 1907, the centrifugal compressor had developed so that it became a commercial product of itself. One early result was the use of centrifugal compressors for blast furnaces for production of pig iron. This met early opposition from advocates of reciprocating "blowing tubs," and it was alleged that a centrifugal compressor, operated by a flow-metering device to deliver a constant weight flow of air, nevertheless was subject to a process called "churning" whereby the impellers would rotate without delivering any air, in spite of the indication of the flow meter. This was shown to be a fallacy. Also it was shown that even though internal-combustion engines operated by blast-furnace gas might have somewhat better thermal efficiency, the small fixed and maintenance charges of the centrifugal compressor gave lower total cost of blast-furnace operation.

Rateau did some early centrifugal-compressor work already mentioned, a great deal of it for gas turbines. Some work probably also was done elsewhere in Europe. The Ingersoll-Rand Company has continued the development in the United States. As one result, centrifugal compressors, started in the United States as a gas-turbine by-product, now are almost exclusively used throughout the world for blast-furnace blowing, as well as for many other purposes.

As another part of the General Electric development, geared centrifugal superchargers for United States aviation engines began about 1925, and now are in enormous production, using impellers based on the original Osborne Reynolds design. But in spite of the fact that some of this centrifugal-compressor development was inspired by the gas turbine, there is present evidence that the centrifugal compressor may be exceeded in efficiency by a comparatively recent competitor, the "axial-flow blower." If so, this blower will be the one used in the ultimate gas turbine. These blowers are used in the Neuchatel gas-turbine power plant and Swiss gas-turbine locomotive. But, on the other hand, they have a shorter operating range and many more stages than the centrifugal compressor, and progress is being made in improvement in efficiency of the latter. However, this is a matter for the future, and cannot be settled here.

At any rate, the gas turbine had its part in introducing centrifugal compressors for blast furnaces, many industrial purposes, and present geared and turbosuperchargers.

United States Gas-turbine Research Continued.

The General Electric gas-turbine work, previously mentioned as beginning in June, 1903, continued until about 1907. At first the power for compression consumed all of the turbine power. Some approximate figures will show why. Suppose a gas-turbine wheel theoretically delivered 100 Hp., and suppose the compressor theoretically required 49 Hp. Then there would be theoretically 51 net Hp. But the turbine wheel might have an efficiency of 70 per cent, so that it actually would deliver 70 Hp.; and the compressor might have an efficiency of 70 per cent, so the net horsepower would come out $100 \times 0.70 - 49/0.70$. Before the reader is cruel enough to laugh at this result, let him put himself in the place of lots of us gas-turbine inventors who have sweated blood through years of research only to come out this way or worse. The reason is that, with the 51 theoretical Hp., net, we have apparatus of 149 theoretical Hp., with consequent losses, instead of the losses directly corresponding to 51 Hp. A reciprocating internal-combustion engine deducts the power for compression from the gross power, before any power is delivered, and so the losses are based upon the net power output rather than upon the sum of powers of compression and expansion.

This article purports to give items other than the vast number of gas-turbine thermodynamic computations which have been published by many persons, the author among them. But a simple approximate computation can be made which will show the other side of the foregoing picture of a gas turbine with a zero net power. This is on the basis that the products of combustion act as a perfect gas and have mass and properties of the same nature as those of the original air for combustion. Suppose the initial pressure for the compressor and the exhaust pressure of the turbine is 14.7 psia and suppose the final pressure of the compressor and the nozzle inlet pressure of the turbine is 49 psig or 63.7 psia. With such equality of initial and final pressures for turbine and compressor, the ratio of the theoretical power from the turbine to the theoretical power required

for the compressor is equal to the ratio of the absolute temperature at nozzle inlet to the theoretical absolute temperature at the end of isentropic air compression.

A test of the Neuchatel gas turbine is cited later, and its figures should be before every gas-turbine student. The ratio of the theoretical powers in test No. 2 is 1.83, and the ratio of the absolute temperatures is 1.82, which shows the character of the approximation here being made. With 70 F. inlet temperature, the compressor isentropic final temperature is 802 F. absolute. Suppose the temperature at the turbine nozzle inlet is 1946 F. This is a high temperature, but within the bounds of possibility. The absolute temperature at the nozzle inlet then is 2406 F. The ratio of absolute temperatures, which is approximately the ratio of the theoretical powers, then is 3; that is, the theoretical power for the compressor is 33 per cent of the theoretical turbine power.

Using the efficiencies of the Neuchatel gas-turbine test, the compressor efficiency is about 85 per cent, so the actual power for the compressor is 0.33/0.85 or 39 per cent of the theoretical turbine power. With 88 per cent efficiency of the turbine, the actual power from the turbine is 88 per cent of the turbine theoretical power. That is to say, the net power is (0.88-0.39)/0.88 or 55 per cent of the turbine actual power. This figure is far beyond the 1907 possibility, and is on the basis of all of the progress since that date, and then some. The Neuchatel gas-turbine test has a turbine temperature of about 1000 F., much lower than the 1946 F. previously assumed, so that the net power only is about 27 per cent of the actual turbine power. But, while in 1907, we hoped for everything that since has been shown by the Neuchatel test, the turbosupercharger, and all the other progress, the actual reality was far different.

So the General Electric gas-turbine research continued until a fuel consumption of 4 pounds of kerosene per net Hp.-hr. was in sight. At that time good oil engines were using 1 pound of oil per net Hp.-hr. Even though it seemed certain that the clouds cast upon the operation of a gas-turbine nozzle, had no basis in fact, it was thought desirable to measure actual spouting velocity of products of combustion. So the combustion chamber was flexibly mounted, and nozzle reaction measured.

The work was difficult to conduct with accuracy, but enough was done to indicate that the only corrections needed were to take account of departures from the perfect gas laws of the products of combustion. Similar nozzle-reaction experiments later were conducted by Glenn B. Warren with similar results. Recently, Dr. Chester W. Smith and other General Electric engineers have worked on tables of available energy of products of combustion for turbosuperchargers, with exact values of specific heat. But probably more work needs to be done to get exact available energies for gas turbines.

All of the early General Electric gas-turbine work was conducted with temperatures possible with materials then known, and with creep unknown. No vision was had of the possibilities which the turbosupercharger work has developed. So the temperatures were kept low by air excess or water injection, both resulting in inefficiencies. No way then seemed open to do better, and so the gas-turbine part of the research was postponed, but with vigorous commercial work on centrifugal compressors, as already mentioned.

Among the many persons who long have had gas-turbine ideas are two with whom the author has had the pleasure of personal acquaintance, and who have made some publication, Dr. Harvey N. Davis, President of Stevens Institute of Technology, and Glenn B. Warren, Designing Engineer of the Steam Turbine Division of the General Electric Company.

Explosive and Continuous Combustion.

The gas turbine had been expected to replace the reciprocating internal-combustion engine, and so there was an early idea that it was essential that gas-turbine combustion should be similar. This led to a number of proposals for explosive gas turbines, as in the case of Le Pontois, already cited. A German engineer, Hans Holzwarth, has made extensive publications on such apparatus through many years. But there seems doubt as to whether he ever reported performance of a complete machine which compressed its own air.

Opposed to these explosive gas turbines are those based on continuous combustion under pressure. The possibility of efficient operation once was doubted. Since then many continuous combustion chambers have been operated with complete success, so demonstration on this point no longer is needed.

An original, or at least an early continuous-combustion chamber, was that of the Doctor's Thesis. Such combustion in a chamber at a pressure of any amount above atmospheric proceeds exactly as it does in the open combustion chamber of a steam boiler, where the absolute pressure is about 14.7 psi; and the same heat of combustion is liberated. Because of constancy of conditions, the nozzle-spouting velocity is constant, which is a necessary condition for efficient operation of a turbine. On the other hand, an explosive combustion chamber delivers a jet with variable velocity, which can be proper for the turbine speed during only a fraction of the time, and so results in inefficiency. At any rate, modern gas turbines are wholly on the continuous-combustion basis, and explosive outfits are not now attracting any attention.

Gas-turbine Prime Movers.

Thus, the gas-turbine idea has been kept alive through many years, with some partial approaches, as will be described later, but always with the hope among enthusiasts that some day it will compete with such high-efficiency prime movers as the modern steam central station or the internal-combustion reciprocating engine. Prof. Lionel S. Marks, now Professor Emeritus of the Graduate School of Engineering of Harvard University, long has been a gas-turbine student, and on December 14, 1939, he gave a lecture which presented a realistic view of immediate gas-turbine prime-mover probabilities. He pointed out that the successful use of the temperatures necessary to give high efficiency might be far off, but that there were many immediate prospects with lower temperatures. Among these were locomotives to compete with the modern steam locomotives of moderate efficiencies, plants where water is scarce, and stand-by plants where low first costs would give total charges competing with more efficient but expensive units. Similar ideas were expressed by Dr. Adolph Meyer of the Brown-Boveri Company, in a paper which mentioned the starting in the Brown-Boveri shops of the Neuchatel 4000-Kw. gas turbine, and the Swiss Federal Railway gas-turbine locomotive, the completion of which are described next.

The town of Neuchatel, Switzerland, arranged for an emergency power plant, in a bomb-proof place, as a war protection, and a 4000-Kw. gas-turbine plant was constructed. An efficiency test of this in actual operation was published by the famous German turbine authority, then in his old age, Dr. A. Stodola, and seems to be a worthwhile operation of a gas-turbine prime mover.

Recently actual test results have been published of the gas-turbine locomotive mentioned by Dr. Meyer as being constructed by Brown-Boveri

for the Swiss Federal Railway, which seems ready for actual use. This is another operation of the sort of gas-turbine prime mover which has been sought through many years. Both of these gas-turbine prime movers are of the sort that Prof. Marks and Dr. Meyer mention, where useful purposes are served by temperatures of about 1000 F. But like the other cycles here mentioned, they pave the way for such higher temperatures as will make high gas-turbine efficiency a primary object. Admiral Mills recently has mentioned development by the United States Navy of gas turbines for ship propulsion.

Partial Gas-turbine Cycles.

There are some gas turbines in extensive commercial operation, but not as complete prime movers. An early one is the gas turbine in extensive use by the Brown-Boveri Company and others, to supply compressed air to the supercharged combustion chambers of steam boilers for the Brown-Boveri Velox System, and other similar systems.

Many partial gas-turbine cycles have been in successful operation for years, for oil refineries, built by the Allis-Chalmers Company. Outfits of a similar sort have been constructed or planned by the General Electric Company and others.

All of the foregoing outfits are planned for the comparatively moderate temperature of about 1000 F. They use gases produced in the course of chemical and combustion operations, and drive air compressors. They do not have the obligation imposed on a gas-turbine prime mover, which has to produce net power as well as to compress the air which produces, by combustion or otherwise, the fluid which drives the turbine. But they are definite steps in gas-turbine progress.

In the meantime, temperatures used in steam turbines have been rising, and many are operating at temperatures around 900 to 1000 F. and show present general use of such temperatures. But to a "red-hot" gas-turbine enthusiast, who lathers himself into a white heat in trying to get white-hot gas-turbines, a mere 900 F. is almost a refrigerator temperature. But it is a step on the way.

Reciprocating Engines Exhausting into Gas Turbines.

A gas-turbine enthusiast dreams of complete prime movers such as the Neuchatel power plant and the gas-turbine locomotive, but with such temperatures as will give efficiencies competitive with any other cycles. However, there are those who argue that, for moderate powers, the upper part of the high-temperature range which is necessary for high prime-mover efficiencies, can best be handled in a reciprocating engine, with the gas turbine to take the gases at a reduced temperature and carry the expansion to the limit. But whether this is going to be the ultimate is a question. There are three sorts of such cycles in present use where reciprocating internal-combustion engines exhaust into gas turbines. These may be called (*a*) Diesel-boiler or Gotaverken-Sulzer cycle, (*b*) Buchi Divided-manifold cycle, and (*c*) constant-pressure-turbosupercharger cycle, and will be considered in this order.

(*a*). *Diesel-Boiler Cycle.* An early system was proposed by the engineer Johannsen of Gotaverken, a prominent shipbuilding concern of Gothenburg, Sweden, of which Hugo Heyman has been Managing-Director. A two-cycle Diesel engine exhausts at the high back pressure of about 4 atm., and drives only its scavenging reciprocating air compressor, which is mounted on its crankshaft. The exhaust, cooled by a

large amount of scavenging air, proceeds to the turbine, all of whose power is net. Sulzer Brothers, Ltd., of Winterthur, Switzerland, has been working on various modifications of the same general plan. Free pistons also have been proposed.

(*b*). *Divided-Manifold Cycle.* Dr. Alfred Buchi of Winterthur, Switzerland, long has worked on turbosuperchargers, and, as described next, made a very early proposal for the modern type with constant pressure, since used by others. Buchi, and his associates, the Brown-Boveri Company, also have used what may be called a divided manifold. This has a Diesel engine with the usual sort of turbosupercharger, next considered, except that nozzle-box sections and exhaust-manifold passages are divided so that a given exhaust passage, into which exhaust is proceeding from one cylinder, is never connected to another cylinder which is at the end of exhaust and beginning of intake. Buchi says there never can be "blow-back" of exhaust gas into the intake manifold during the overlap period when both exhaust and intake valves of a cylinder are open at the same time. This cycle also has been used by Rateau.

In most cases, the cycle is used with Diesel engines having low exhaust temperatures and low speeds, as compared with aviation engines. The divided-manifold sections are of such reduced area as to avoid sudden drop of exhaust pressure. Furthermore, each cylinder exhausts by itself until its exhaust pressure dies out. So there is a variable nozzle-spouting velocity, which does not match the turbine-wheel speed during the entire time, and thus causes a loss. The system of interlocking sections of the exhaust manifold connected to the different cylinders causes appreciable complexity.

With a turbosupercharger system of such inefficiency that the intake-manifold pressure is nearly equal to or less than the exhaust-manifold pressure, the divided-manifold-system advocates say that it avoids "blow-back" and gives scavenging instead, and so maintains full engine power. But with the aviation turbosuperchargers next described, the efficiency is such that the intake-manifold pressure is higher than the exhaust-manifold pressure, and so there is proper scavenging with the constant-pressure-turbosupercharger cycle.

History of Aviation Turbosuperchargers.

The constant-pressure-turbosupercharger cycle first began practical use with high-altitude aviation. The constant pressure started as that of sea-level atmosphere, 30 inches of mercury absolute, and the expansion and compression are to and from the low pressure of altitude atmosphere. The part which this aviation turbosupercharger has in the gas-turbine picture will be begun with its history, by taking a flashback to World War I. Military aviation was in its infancy, but already there was difficulty in the loss of engine power even at the moderate altitudes then possible. Hence, efforts were made to attach, by gearing, air compressors of various sorts, to give airplane-engine supercharging, but none was successful. General Electric later supplied geared superchargers having the design of the centrifugal compressor developed during the gas-turbine research, but these did not come into use until 1925.

During World War I, Dr. William F. Durand was Chairman of the National Advisory Committee for Aeronautics, and in the course of one of his duties of examination of schemes for aviation improvement there came to him, in the fall of 1917, a plan of the French engineer, Rateau, for using a turbine wheel driven by the products of combustion of the engine exhaust, at constant pressure, to drive a centrifugal compressor to

do the supercharging. Dr. Durand had been a professor at Sibley College, Cornell University, in 1901, and he remembered the black smoke from the gas-turbine research, and knew of the extensive business that the General Electric Company had in steam turbines and centrifugal compressors. So he asked General Electric to undertake turbosupercharger development in the United States. General Electric engineers had their own ideas about design details from their experience with gas turbines, steam turbines, and centrifugal compressors, and so started a development entirely independent of Rateau, which has continued to this day, with production successively increasing on account of the present war.

The first General Electric turbosupercharger was constructed with wartime speed, and was assembled on one of the Liberty motors then being used by the U. S. Army Air Corps, McCook Field, Dayton, Ohio, in May, 1918. It was operated there so far as possible at sea level, and then tested at the 14,000-foot summit of Pikes Peak (Colo.).

The airplane turbosupercharger is so arranged that, as the plane ascends to the successively lower atmospheric pressures at altitude, normal sea-level pressure continues to be maintained in the exhaust manifold and turbine nozzle box. The difference between this and the low pressure of altitude gives the nozzle jet that drives the turbine wheel. This drives a centrifugal-compressor impeller on the same shaft, which compresses the low-pressure air of altitude and supplies it at sea-level pressure to the engine intake manifold. So, regardless of altitude, the engine exhausts at sea-level pressure and receives its charge at sea-level pressure. So it continues to deliver sea-level power at all altitudes, up to the critical one where the turbine wheel reaches its maximum safe speed. Due to the low resistance of the low-density air at altitude, sea-level engine power gives increased airplane speed.

The constant pressure of the engine intake manifolds of the early airplane turbosupercharger was 30 inches of mercury. This is greatly exceeded nowadays. Values around 50 inches are usual for takeoff or short periods, and continuous operation is common with "boost" of intake manifold pressure as referred to later.

Rateau's turbosupercharger development in France continued for a while after World War I, and developments were started in England and Germany. But the resources devoted to the project abroad do not seem to have been sufficient to bring it to a state that warranted continuation. At any rate, there is no evidence of any turbosuperchargers other than those of General Electric being used in the present conflict. Other differing American designs recently have begun to be developed.

It once was thought that the 1917 work in France was the first mention of the modern turbosupercharger plan, but later searches show what seems to be an earlier publication by Buchi. Nevertheless, Rateau made important contributions.

In the United States, the development encountered the usual experience of the "glassy eye." But there were enough enthusiasts among the officers and engineers of the United States Army Air Forces, and enough help from them, from engine and airplane manufacturers, and from metallurgists, and sufficient resources available to General Electric, to bring the project to its present undreamed-of state.

The principal interest that the turbosupercharger has in connection with the gas-turbine problem is its development of a turbine to operate at high temperatures of products of combustion. It is true that the aviation turbosupercharger has only short-time operation at extreme conditions, and does not have to meet the conditions of continuous operation in a power plant. Nevertheless, many problems have been solved, due to the impetus of

World War II, which advance the prospects of a gas turbine as a prime mover. Thus, the turbosupercharger, originally a gas-turbine by-product, has not only proved very useful of itself, but has accelerated gas-turbine development.

The turbosupercharger in the United States started in 1918, with a turbine wheel suited only for steam temperature, provided in a rush by the De Laval Steam Turbine Company. Since then there has been continuous progress in materials, fabrication, heat-treating, design, and cooling, of all of the parts, whether exposed to exhaust-gas temperatures or not.

Originally the nozzle area covered only a portion of the wheel periphery, with cooling between, and this may be an item in the ultimate gas turbine. But for many years the turbosupercharger has had a 360-degree nozzle arc. For years, the rated temperature at the nozzle-box entrance was 1200 F. This meant that the entire nozzle box and the outer portion of the turbine buckets were visibly red hot, at rotative speeds of about 22,000 Rpm. Successive improvements have permitted much greater rated temperatures. Since about 1930, a part of the General Electric turbosupercharger research has been a "hot-gas test" where turbos were operated in the laboratory on products of combustion, with all conditions as when flying at altitude. By opening a peekhole in the altitude chamber in which the turbine wheel was running, there was seen for the first time operation with red-hot turbine buckets, and colored photographs were taken. Since the author always has considered the turbosupercharger as merely a step on the way to the gas turbine, he has often exhibited as gas-turbine propaganda this really inspiring sight of a turbine wheel operating at about 1200 feet per second with buckets red hot.

(*c*). Constant-pressure Turbosupercharger Cycle.

Turbosuperchargers on American airplanes are examples of the third of the cycles previously mentioned, where reciprocating engines exhaust into gas turbines. They are designed for pressure ratios which give absolute pressures above sea-level value, when flying at altitude. This means that absolute pressures in the engine-intake manifolds appreciably above sea-level values are easily possible at moderate altitudes. Such pressures are used for take-off or for regular operation under special circumstances, which is called "boost." Pilots flying airplanes with turbosuperchargers have instruments showing airplane altitude, intake-manifold pressure, turbosupercharger speed, various temperatures, and so on. They have rules and technical orders which give values for the readings of the various instruments, which give values for the "boost" of engine-intake-manifold pressures safe for the engine and turbosupercharger. The greater the boost, the greater the engine horsepower and the greater the airplane speed.

Thus, when a pilot with a turbosupercharger is in battle, he has means of greatly increasing his plane speeds at most altitudes, by increased boost. So, when an enemy plane is on his tail, does the pilot take out his technical orders, look at his instruments, and figure out what he ought to do? He does *not*, but subjects the turbosupercharger to pressures far beyond its test rating.

Extension of such operation, which easily is possible independent of aviation, would give a four-stroke Otto- or Diesel-cycle engine with a constant-pressure turbosupercharger, which is this third one of the cycles where a reciprocating internal-combustion engine exhausts into a gas turbine. It has been called a "compound internal-combustion cycle."

As already mentioned, reports of operation of turbosuperchargers in war show that they are subjected to conditions far exceeding their test rating. Therefore, the average of the conditions in war has some relation to conditions of long-time operation at sea level to which a gas-turbine prime mover would be subjected.

High-temperature Progress.

One major item in operation of turbine wheels at high temperature is "creep," which is continuous nonelastic elongation, finally resulting in rupture. Hence, stresses for a gas turbine to be operated in a power plant for years, are lower than for a turbosupercharger in war. Information about creep is rapidly accumulating in connection with turbosuperchargers as well as with other turbine work.

Reference already has been made to Buchi, Brown-Boveri, and Rateau work on turbosuperchargers with divided manifolds probably always with low exhaust temperature, low-speed Diesel engines. Accounts also have been published of Swiss, French, and German use of turbosuperchargers on aviation engines, with Diesel or Otto cycles and with divided or constant-pressure manifolds. None of the apparatus thus referred to is in use in World War II, the only known turbosuperchargers being those of the United States Army Air Forces, which as already mentioned are in very extensive use.

Nevertheless, some progress is being made abroad with turbine wheels operated by products of combustion. One example is a publication by Brown-Boveri, which includes a night picture of a turbosupercharger for an aviation engine (probably with a constant-pressure exhaust manifold) in a factory test at about 1800 F. and 30,000 Rpm. Glowing exhaust-gas conduit and nozzle box are shown with the remark, "The set presents an exceptional and fascinating appearance. It is surrounded by the glowing gas pipe and carries the mind back to some medieval machine."

Thus, the thousands of turbosupercharger turbines, in operation by the United States Army Air Forces, and being investigated abroad, are adding to the other gas-turbine items already mentioned, in paving the way for temperatures much higher than those which have been used, and with a minimum of loss from cooling. No thermodynamic studies can be given here, but it is well understood that such higher temperatures mean higher gas-turbine efficiencies. So the long-sought gas-turbine prime mover, with temperatures much beyond the 1000 F. now in use, and consequent higher efficiency, seems to be getting nearer. Of course, this same progress is paving the way for higher temperatures and higher efficiencies with the regular steam cycle and with two-fluid cycles. So we only can speculate as to whether the gas turbine with its compressor, combustion chamber, and heat interchanger is going to compete with the other cycles which instead have economizers, feedwater heaters, feed and air pumps, boilers, and condensers.

PRODUCTION REQUIREMENTS AND ENGINE DESIGN.

Mr. R. E. Strub, a well known Swiss Diesel engine specialist, one time Diesel engine designer for W. H. Allen, Sons and Company, Ltd., later with British Auxiliaries, Ltd., and currently with Samuel Hodge and Com-

pany, London, presented the following paper at a recent meeting of the North-East Coast Institution of Engineers and Shipbuilders in Newcastle-on-Tyne. His paper, abstracted here, is principally of value in demonstrating the importance of close cooperation between designer and engine builder. The paper is largely concerned with heavy-oil engines of the medium power medium-speed type.

When viewing the field of application of reciprocating machinery, principally the internal-combustion engine for marine propulsion, it is a fact worthy of note that many shipowners when laying their plans now for their program of construction of new tonnage after the war, are investigating the possibility of adopting "indirect" propulsion, either geared or Diesel-electric. The motive behind the quest for prime-mover units smaller than the present-time directly driven propulsion sets is not very difficult to find. It may well be that in a program comprising a number of ships going into two figures, the tonnage and the cubic capacity of an additional ship can be gained without additional expenditure, thanks to the economy of space and weight on the ships equipped with "indirect" drives. It is contended that this advantage will be obtained even when considering the increased bunker capacity necessitated by the drop in efficiency in Diesel-electric drives, and even more in geared drives with their smaller losses. If these contentions are accepted, the propositions put forward for indirect drives will also have to be examined on the grounds of their reliability in service. Not only can a saving in weight and space be achieved, but the initial cost of the prime mover itself can be reduced because its decreased size lends itself to more economic production.

Engine Production.

Factors which will influence the outlook of the production policy are the finer tolerances and the production of greater numbers of smaller parts than when compared with the 160-pound category. The weights of the engine parts are much lower and therefore can be more easily handled. They are of such a size that in many cases replacement can be obtained when necessary by the engineering facilities available in a large number of ports instead of only from a few engine-builders in the foremost marine-engineering countries. With modern workshop practice, the smaller parts are made with greater precision, thus expediting assembly speed. The works organization will be affected by the adoption of modern tooling borrowed from mass-production methods, and by the application of a scientific basis for the fixing of speeds and feeds of the machine tool, and by modern systems of planning and progressing by loading of the machine tools. Probably the change in the workshops will be more pronounced when changing from the slow-speed to the high-speed oil-engine than it was some twenty years ago at the time of the transition from steam-engine to slow-speed Diesel engine. The change will be likely to show its effects in a general speed-up in the engining of the ships. This is a point worth remembering if one should be thinking in terms of a wholesale renewal of a fleet.

The numbers of repetitive parts to be produced, however, do not approach those warranting mass-production methods proper, *e.g.*, the exclusive use of the single-purpose machine tool, but the numbers involved do call for the adaptation of general-purpose machines, and for special precautions in design of the power units to lend themselves to these adaptations.

In this connection it is interesting to note the methods adopted by the Production Section of the U.S.A. War Department when originating a large-scale campaign with a view to covering the no-man's-land between designer and producer in ordnance work. A number of booklets appearing under the appropriate title of "Tremendous Trifles" were published by that Department showing numerous examples indicating how, by slight changes in design of various items, simpler manufacturing methods and more generally available materials can be substituted for the methods and materials formerly used. A few of these examples indicate the great savings possible through such changes, and suggest where similar alterations could be made in other equipment, stimulating thought along simplification lines that will speed up production and reduce costs without any change in the quality of the product.

To return to our subject, it will be realized that no hard and fast formula can be given by which these "trifling" changes in design can be effected to give these, if not "tremendous," yet certainly beneficial results.

A selection of concrete typical examples of machining operations such as turning, boring, planing, drilling, and welding of reciprocating engine parts will be given in the following pages.

Examples of Alternatives of Design.

The boring of the bedplate for a multi-cylinder engine is one of the operations liable to index the output of a machine shop. Most works will not have many machines at their disposal to provide the heavy drive required, and the volume of output will probably be bottle-necked round the largest Kearns or Richards type horizontal boring machine.

A typical example of a case in which the designer has not considered the effect of a particular detail on production requirements is shown in Figure 1. The summation of the bearing loads on the forward-end bearing

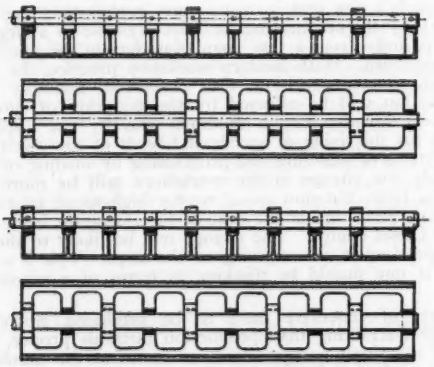

Fig. 1

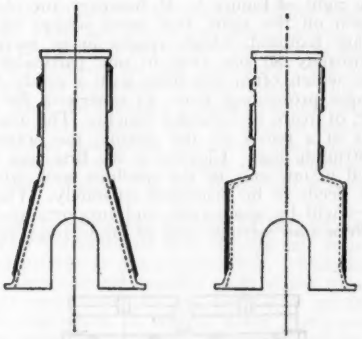

Fig. 5

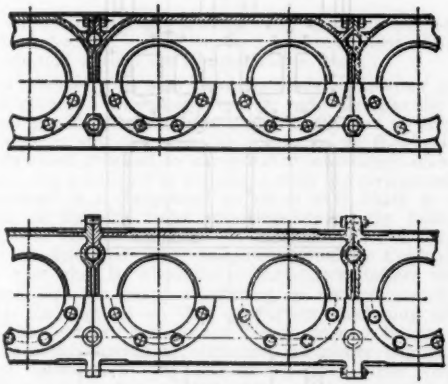

Fig. 6

outside the drive for the engine accessories, shows it to be stressed to a much smaller degree than the main bearings adjacent to the working cylinders. Consequently, the designer provided for a bearing with an appreciably smaller diameter. This smaller diameter calls for a smaller boring bar which, without excessive deflection and torsion, may not take more than two tools simultaneously. The boring operations consequently have to be carried out in five settings of the tools and steadies.

The bedplate at the bottom of Figure 1 shows the same diameter for the bearing keeps throughout, and a special heavy boring bar can be provided which will exploit to the full the power of a heavy machine tool, and each of four boring operations can be completed in two settings.

Another example in which the number of set-ups can be decreased is shown on Figure 5 dealing with the lateral milling or planing of the cylinder blocks. A large planomiller of the Ingersoll type can mill the sides of the blocks straight through in one setting if they are vertical,

as shown on the right of Figure 5. If, however, the plane of the frame is inclined as shown on the right, two more set-ups of the frame on the machine table are required, which results in an increase of production time of approximately 80 per cent in one particular case investigated.

Other features which often interferes with a steady flow of production are lugs or flanges protruding from an otherwise flat surface. Figure 6 shows the joint of pairs of cylinder blocks. The top half of the slide shows the joint in a recess of the casting, the bottom half shows the design with protruding lugs. Clearly, in the first case two or more pairs can be machined in one run of the machine tool, whereas in the latter case each block needs to be machined separately. The effect of this on production times will be appreciable and proportionate to the additional number of machine-tool settings and of measurings required.

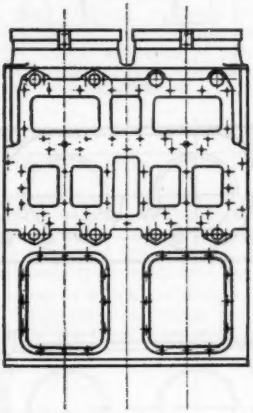

Fig. 7

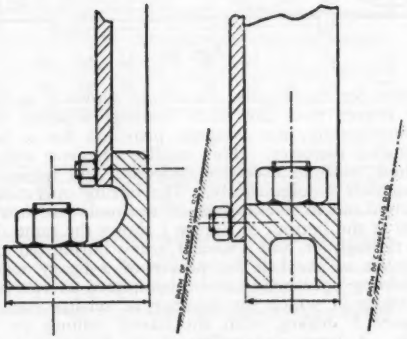

Fig. 8

Reverting now to examples comprising drilling operations, two halves of a twin-cylinder block are shown in Figure 7 which at first sight appear alike. However, the designer has spoiled the flow of drilling operations on the alternative half shown on the right-hand side by embodying two threaded holes for the insertion of core plugs. This design necessitates shifting the job from the ordinary drilling machine which may deal with holes up to two inches diameter only to another machine. The arrangement on the left-hand side shows the cored holes themselves unmachined, but covered by flanges attached by studs in the ordinary way. The time saved by this slight change will be about half-an-hour per block in a particular case.

Alternative methods of drilling the foot of the cylinder block are shown on Figure 8. The conventional way on the left calls for an additional milled or planed surface at the extreme left, which latter is necessitated for purposes of alignment on machine tools at the subsequent set-ups for boring and other operations. In the alternative on the right, only one surface has to be machined. It will be noted that a saving in weight and space is obtained affecting the overall dimensions of the bedplate which is fitted below the cylinder block, apart from giving a compact and neat appearance.

Figure 9 shows the saving of weight and space by the adoption of the so-called "socket-screw." This method of attachment shown is said to have been adopted abroad on an appreciable scale.

As a final example, two alternative methods of flywheel attachment are shown on Figure 10. The design on the right obviates the machining of keyways and produces an additional lightening due to the reduction of the boss, which in any case has only a very minor flywheel effect.

With a flywheel attached to a crankshaft of 12-inch diameter, a saving of two hours was calculated as compared with the arrangement on the left.

The crankshaft is a component which is very likely to index output. Power units of the size under consideration usually have solid crankshafts, and thus the manufacturer is in the hands of the forge-master. The shafts are delivered either rough-machined or finished-machined. In the former case, that is, when the manufacturer finishes the crankshafts, the tolerances for the rough machining of the crank-webs can be arranged with the supplier so that no further machining of the webs is required. The primary unbalanced forces created by the coarse tolerances of the rough-machining operation are either negligible or can be compensated satisfactorily by adjustment after static balancing. New progress, however, is being made through the adoption of composite crankshafts. One of the most promising of the alternative designs for the size of the engine under consideration is the one in which the crank-pin is solid with the crank-webs. This part can either be made in cast steel, or as a forging by the facilities of a number of shipyards or works. The journals which are shrunk into the webs can be hardened and ground, which is an appreciable advantage for the higher-speed engine because the harder bearing material made possible by the harder journal will stand higher specific bearing pressures. Inversely, if the designed specific bearing pressures are left low, although the bearing is capable of higher loads, the wear will be kept to a minimum. This is of special importance to the higher-speed engine because, due to the larger inertia forces, the specific bearing pressures vary widely all along the engine, owing to the varying crank settings. Wherever the adjoining crank throws are in opposition, inertia loads compensate and the bearing pressure is low. If, on the other hand, the crank-pins are disposed symmetrically, the inertia loads add. Consequent uneven wear of the journal bearings will endanger the crankshaft more quickly than almost any other cause.

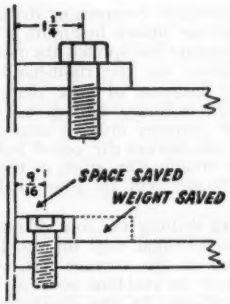

Fig. 9

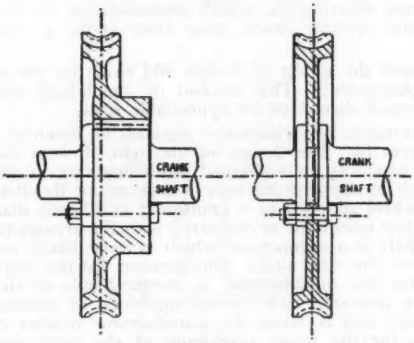

Fig. 10

All crankshafts are today designed for torsional rigidity so as to increase the natural frequencies of vibration for the multi-cylinder engine, and the diameters of the journals have become comparatively much larger on the modern engine. This feature will contribute in providing a good shrinking surface for the composite shaft. True, the torsional frequencies are somewhat lowered by the additional rotating mass of the webs gripping the journal, but this can be compensated partly by removing metal on the end of the web where it joins the crank-pin. This can be done without impairing the rigidity by molding the webs to the best stressing advantage, and any deduction of material on the outer end is effective due to its greater radius of gyration.

The shrinking operation requires particular attention. The crank-pins and the journals are not only to be set at accurate angles relative to each other, but have to keep in line when the crankshaft has cooled down. Of several methods for obtaining this the following is suggested:—The crank-pins are made hollow and are provided with an accurately machined bore which is true centric and parallel with the outer surface of the pin and therefore also with the holes into which the journal pins are shrunk. The crankshaft is assembled in a vertical position by fitting the bores of the crank-pins over the fixture consisting of vertical columns

having the same diameter as the holes in the crank-pins over which the latter are slipped. Figure 11 shows in its top half the conventional design of a crank element which has to be machined all over. In the lower half of the same figure a crank element is shown with journals shrunk into the cast web. An estimate of the machining operations on a 9-inch diameter shaft shows that the former design needs approximately twice the time for manufacture as compared with the cast-steel design.

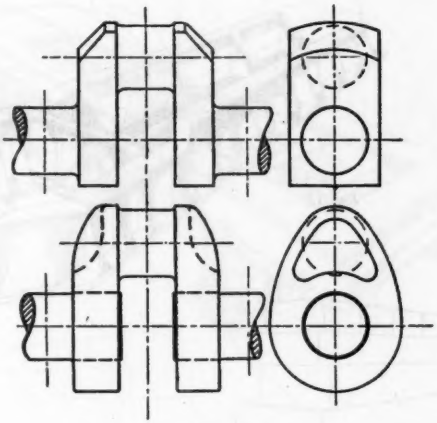

Fig. 11

Welding.

With welding as it is developed today, particular consideration must be given to fatigue stressing when considering fabrication of main structures for the type of engine in view. Fabricated combination base-plates, however, have become established. Figures 12, 13, 14 and 15 show some of the progressive development over the last years. Figure 12 is a typical example of what welded design should *not* be. No welding engineer will be surprised that a base-plate which was actually made to this design, and which is but a copy of previously made castings, cracked in a number of places on its way from the welding shop to the test bed, and in spite of re-welding and heat treatment found an early end on the scrap heap. Figure 13, which utilizes drawn profiles and avoids butt-welding in critical places, is an improvement, and base-plates to this design have given rise to no complaints. In Figure 14, which shows an elastic foundation, further regard has been given to the requirements of the welder. The beams running transversely are reposing on the flanges of the longitudinal joists and are welded at the junction at the bottom and underneath, thus avoiding the butt-weld. From the point of view of noise prevention, this base could be improved by dispensing with the through-bolts going from the bottom of the flange to the ship's foundations. Even with rubber bushes around these bolts it is difficult, if not impossible, to avoid close contact between the elastic suspended system and the foundations. Figure 15 shows the latest development consisting of an outer and an inner

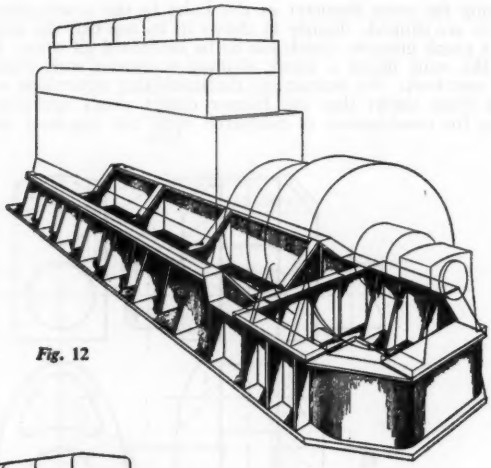

Fig. 12

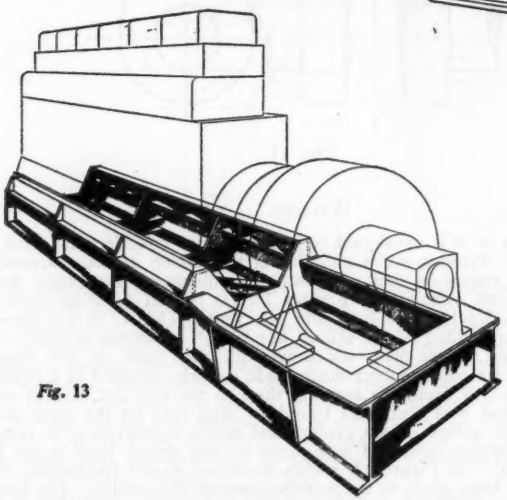

Fig. 13

channel bent round at the generator end, and held together by top and bottom plates. A similar channel runs across the free end of the base. The structure as a whole is further stiffened by providing bent ends at the generator end. It is these corners which support the pedestal bearing and which are indexing the strength of the whole of the structure. As to the bending of these channels, they present no difficulty to a ship-building yard.

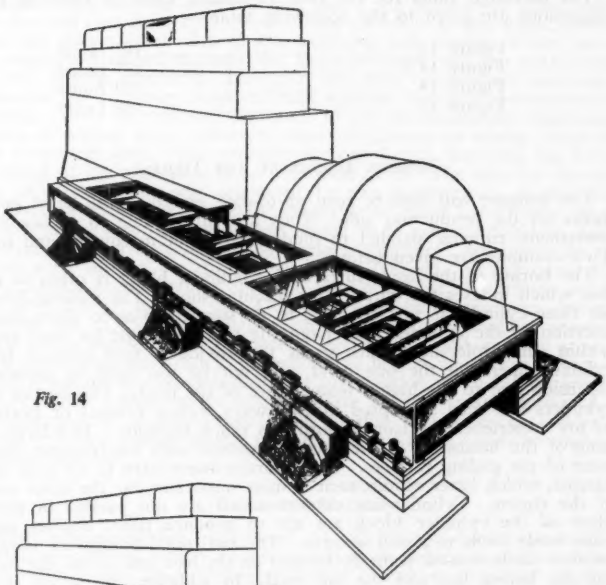

Fig. 14

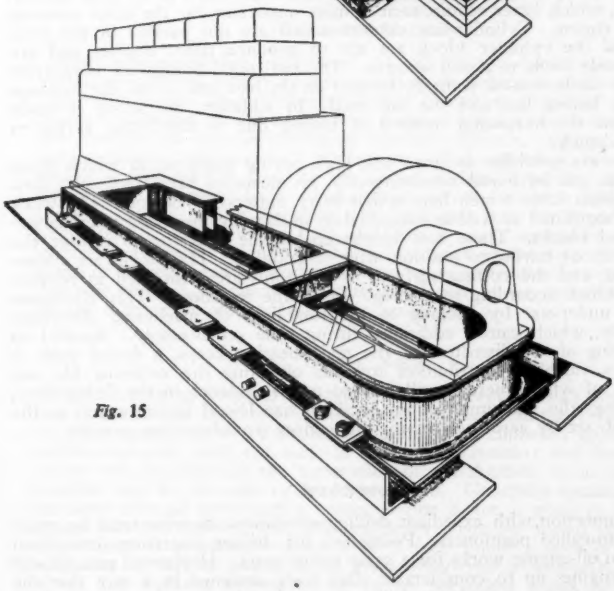

Fig. 15

The estimated times for the four fabricated bases of identical overall dimensions are given in the following table:—

Figure 12	197 hours
Figure 13	155 hours
Figure 14	139 hours
Figure 15	108 hours

Modern Machinery and Tooling.

The designer will have to keep up-to-date regarding the latest developments on the production side. Thus he will be able to decide on improvements running parallel to modernization in machinery and tooling. Two examples are given below.

The boring of the inside surface of a cylinder block is often an operation which indexes the output of a machine shop. The various diameters for these cylindrical surfaces have to be stepped so as to allow the easy insertion of the cylinder liner from the top, and must be very accurate within small tolerance limits, so as to provide a firm support for the cylinder liners on the one hand, and not be too tight to permit heat expansion due to the higher temperature of the liner. The boring of the cylinders has been attempted in two ways, either vertical or horizontal. In my experience, horizontal boring is much superior. The large overhang of the headstock of the vertical machine calls for frequent readjustment of the gliding surfaces. The cylinder bores have to be kept strictly parallel, which latter requirement is most important for the quiet running of the engine. Cylinder-line centers which are not parallel in the axial plane of the cylinder block are apt to produce noisy engines and are more easily liable to piston seizures. The horizontal boring bar is superior because the headstock is firmly located on the base and so are the bearings for the boring bar and the job itself. In addition, the set-up is made easier on the horizontal method of boring due to the better access to the headstock.

There are specially designed and built boring machines in which three cylinders can be bored simultaneously by means of three hydraulic feed units which drive 6-inch bars which carry cutters for boring. The work can be expedited as well as improved in quality by means of interchangeable tool blocks. These tool blocks enable the operator to change the tools without having to position after re-grinding. The blocks are always centered, and the cutting surfaces of the tools are positioned in relation to the block according to a gauge set in the toolroom. This eliminates boring under-size by keeping to the low limit of machining allowance generally, which causes additional fitting time on assembly. Applied to the boring of the diameters of the main bearing keeps, a device such as this is a valuable contribution towards obviating the ominous file and scraper, of which there is still far too much evidence in the fitting shop, and brings the machining of the medium size Diesel engine nearer to the standards set by aero- and motor-car-engines manufacturing practice.

Positioners.

In connection with expedient drilling operations, mention must be made of the so-called positioners. Positioners for drilling operations have been in use in oil-engine works for a great many years. Horizontal gas, oil and steam engines up to considerable sizes were designed in a way that the bottom surface required no drilling operation from underneath its base

surface. The job was then clamped on to the positioner, which offered the top and four sides to the drill. The accuracy of this work, however, was not always ideal, due to the overhang and weight of the job and the deflection caused through the large leverages and the heavy thrusts of the drilling operation. However, the lack of accuracy was not so important in stationary single-cylinder engines to which this method was applied. With the advent, however, of the multi-cylinder higher-speed engine requiring great accuracy, these positioners or tilting tables were less frequently used. The necessity for expediency, however, has brought a revival of the positioner in an improved form. The idea for the necessity of a positioner was originated by the requirements in the set-up for welding operations that there should be a "down hand" weld on each surface of the job. The ease with which these castings could be held in an exact position appealed to the works management, who gave the positioner a try-out on the drilling machine. Through its improved design good results were obtained, and the positioners for the drilling machine are now highly spoken of. They are now built into sizes having a capacity up to 15 tons. They swing large heavy assemblies around as easily as one can turn a small piece of steel in the hand, and are grounded on a single pedestal which is adjustable for height, and has a table which revolves completely around, tilting to any position up to 135 degrees with the horizontal either up or down. These positioners are motorized and are operated by a push-button panel so that the driller or welder can swing and tilt the work to any position without using cranes or employing a handling crew.

Broaching Machine.

New possibilities for production are offered by developments of the broaching machine, which is now applied to machining of parts with much larger dimensions than formerly. Inside broaching machines generating square or hexagon or any odd size or shape of holes are well known, but a new method of outside broaching converts difficult, tedious, jobs into fast, accurate, "one-machine" operations. This procedure has been extended within the last years to the outside-surface broaching of surfaces which have formerly been produced in several machine tools such as shapers, millers, boring machines, etc., and can now be broached on a single machine in one set-up, with one, two or three operations, according to size and shape. Outside contours of engine connecting rods or the keeps of bearings can be accurately and more expeditiously machined by this new method, provided the great initial cost of the broach is justified by the number of components to be machined.

Engine Noise.

After reviewing some of the relationships between design and production, it may be of interest to study one of its effects on the product itself, *i.e.*, the engine created by this relationship. The improved quality of the engine expresses itself not only in dependable running and low maintenance cost, but also, in the experience of the Author, in a particularly tangible way by its noise or loudness record. Generally speaking, within the same class of machinery a noisier engine is not as good as one registering a lower sound level. Measurements made by means of a sound level meter at a standard distance of 2 feet round a number of ships' auxiliaries of various makes, reveal great differences in intensity, *i.e.*, from 80 to 110 decibels, corresponding to an equivalent range in terms of loudness from fairly loud radio music to the immediate vicinity of a

pneumatic drill. Another feature revealed by these measurements is that the higher speed engine is by no means necessarily the noisier one. The volume of noise, with its corresponding degree of discomfort to the ship's engineers, is of increased importance in multiple-engine installations intended to be serviced by preventive maintenance. A loudness level corresponding to 100 decibels and above, *i.e.*, a noisy tube train, would probably seriously interfere in the long run with the efficiency of systematically organized overhauls in rotation of the power units.

Draughtsmen and Works Practice.

Reviewing, in conclusion, recent developments and their importance for the future, the need must be stressed for giving the designer and the draughtsman every facility and encouragement for making themselves acquainted with the capacities and the potentialities not only of the principal machine tools used in the building of the engine, but also of the procedures and the workshop practice of the works which have to carry out their instructions. This need becomes all the more urgent with the class of engines under consideration, due to its manufacturing methods approaching those of mass production. One practical and also practicable way to carry this object into effect is to let each designing draftsman—in rotation—spend a span of time in the jig and tool drawing office. Assuming the full support of the works management, he will emerge after such a time as a more valuable collaborator to his firm. He will henceforward visualize the jigs and the tools used to produce the part he is setting on paper.

A term at supervision of or stand-by at erection is bound to create a more thorough understanding of the procedure in connection with the important details of foundation and of pipe work, while one or more voyages keeping watches will help to emphasize the necessity for absolute dependability and for accessibility of the engine which cannot be overemphasized.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

INDUSTRIAL INVENTIVENESS.—This most interesting paper on the psychological factors favoring industrial inventiveness was contributed by the Committee on Education and Training for the Industries and presented at the November 29-December 3, 1943, Annual Meeting of the American Society of Mechanical Engineers. It is reprinted from the March, 1944, issue of Mechanical Engineering. The author, Elliot Dunlap Smith, is Master of Saybrook College, Yale University.

The inventiveness needed in industry today is more than "Yankee ingenuity." The old homespun resourcefulness is still as valuable as ever, but to be useful in modern industry the inventor must be both intuitively resourceful and scientifically sound. An example will make clearer what this combination means.

Greeting cards for generations had been printed from steel dies with linseed-oil inks. Then a trend in style made it necessary to have deeper cuts in the dies to make more dramatic effects. This meant that, when the card was printed, the ink, instead of being thinly spread, was massed on the card in thick ridges. Since linseed-oil ink dries by oxidation from the air, the drying of the outer surface of ink created a skin which made

the penetration of oxygen from the air to the ink inside these ridges extremely slow.

To speed up the drying, mechanical engineers devised various schemes, utilizing heat and draft, but without solving the problem. Each color that was printed still took several days to dry. Then an invention was made. Instead of working upon the problem of air-drying at all, a young man conceived the idea of using inks which would dry internally through chemical reaction and, in a short time, an ink base was developed that dried almost instantaneously regardless of the size of the ridge. As a result the cards could go on a belt directly from press to press.

This problem was clearly a technological one and the inventor had to be scientifically competent in order to deal with it. Yet the act of inventiveness which achieved the solution was not logical scientific thought at all. It was a flash of intuition which showed the problem in a new light.

Intuition and Logic Both Essential.

Such intuitive insight, even though scientific, differs sharply from the systematic, explicit process of logical scientific thought. The logical thinker consciously works out his solution. He does so step by step in orderly sequence. He thinks principally in words or symbols, and he can put his logic into language so that others can follow it. The intuitive thinker gets his answer unconsciously by all-at-once insight into the problem as a whole. He cannot put his process of thought into words for he is unaware of the means by which his insight came. Today the inventor must be able to do both types of thinking.

The process of creative insight has much in common with the intuitive formation of images in customary thought. For example, if a curve like this) is displayed with the statement that it is part of a whole the rest of which is concealed, most people will see the whole in their mind's eye as a circle. But some will probably see it as a segment as in Figure 1 (a), or as a sector, Figure 1 (b). If, however, some of the group have been working with lenses, they might see it as one of the following lens forms, Figure 1 (c), (d), or (e).

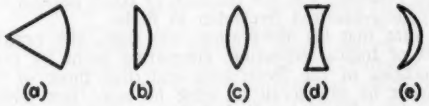

FIG. 1

Whatever the conception, rarely will it result from logical deduction. Of those who saw a circle, probably none will have plotted perpendiculars to tangents and found that they met at a point, and then having found that every point on the curve was equidistant from this focus concluded that the figure was a circle. Instead, the hint provided by the fragment caused each person to see some figure as the whole, without conscious reflection.

It is important to realize that this type of unconscious all-at-once identification permeates our daily life, and that without it we would be helpless. It is important also to realize that a single hint or clue, depending upon the angle of approach, may give rise to a wide variety of conceptions even in a single mind. This is true even when, as here, the element

giving the hint is mathematically precise. Moreover, if one conception fails to satisfy, the mind can shift quickly to another.

What an inventor must do is similar to this. He must resourcefully create out of obscure clues vivid and varied configurations. If configurations of one sort fail, he must be able to view the problem from a completely different angle, such as viewing the curve fragment shown, from the point of view of lenses in case a geometrical approach fails.

Curiously, what the mind recognizes with either hints or wholes, is not the exact patterns which are formed on the eyeball but their structural form. For example, if one holds a circle first at 10 and then at 11 inches from one's eyes, the different-sized images made on the eyeball can have in common only one point of tangency or two points of intersection. Nothing more! And yet at any distance where the image is clear, and even if the sheet is viewed at an angle, no one has difficulty in seeing that the form is circular. This is even true with such complex patterns as chairs, or a person's face. Thus a child has no difficulty in recognizing its mother at different distances or angles.

Intuitive recognition occurs even when there is no common external stimulus; merely different instances of variable structural form or type. For example, there are almost no characteristics common to all Jews. Yet people can ordinarily intuitively recognize a Jew, no matter in what ways he varies from type. This is of fundamental importance because it makes possible intuitive thought in regard to generalizations and abstractions. It thus enables intuition to deal with problems of abstract science.

Applying Intuition Scientifically.

There is no fundamental difference in intuitive recognition when scientific knowledge is involved. Two men, for instance, are walking in the woods. There is a flash of lighter green in the trees. One man sees nothing but a flash of lighter green. The other says: "That's interesting. I have never before seen a yellow-breasted flycatcher in this part of the world." In his well-trained mind, the hints of changing color had been unconsciously associated with what he knows of the bodily structure, way of flight, and color patterns of birds, *into a meaningful whole*, so that what he actually observed was not the flash of color noticed by his friend, but an actual yellow-breasted flycatcher in flight.

If he is not sure that his observation was right, the ornithologist will begin a process of logical reflection, comparing point by point first the family characteristics of the flycatchers and then those of the different species, with what he can recall of what he saw. But what made him aware of the bird at all was not logical thought but the creation, through the unconscious aid of scientific skill, of the configuration of a yellow-breasted fly-catcher out of clues that to the other man were not only meaningless but not even fully observed.

The point I wish to stress here is that, in scientific identification where no invention is involved, what the scientific knowledge first does is to take the same hints which to other men have no meaning and intuitively complete them into a meaningful whole. Even though the process is scientifically exact, the intuitive element is creative.

In invention this creative aspect assumes predominant importance. Unless the inventor is willing to relax the meticulous step-by-step procedure of logical science and let scientific clues suggest generous, or even extravagant, technological conceptions, he will get nowhere. For what the intuitive process of invention does is to lift reason out of the established paths of technology, when they have become ruts, and to put the mind upon a new road leading more directly to its goal.

It is thus when the various habitual images, arising from the clues given by experience, fail to give satisfactory results, that the problem shifts from one of recognition to one of inventiveness. How this occurs is well illustrated by Professor Kohler's studies of problem solving by anthropoid apes.* In one of his experiments, he placed an ape in a cage where food was temptingly near but out of its reach. The ape immediately applied its former practice of reaching through the bars. This failed. It tried the habitual method again and again without result. Then it sat down in discouragement. But the food was tempting. Suddenly it picked up a stick which had been in sight all along and which it had never before used as a retrieving implement. Without any awkward trial and error, without any preliminary steps, it reached out with the stick beyond the food and brought it in.

As with recognition, this intuition consisted in instilling meaning into otherwise unrelated clues. But it further involved creating, out of formerly unrelated hints, a conception which not only fitted the situation but which the ape had never conceived of before. It is in this creation of novelty by intuitive contrivance that the essence of invention lies.

Now it is interesting that during all the preliminary struggle the stick had been in plain sight, but had had no significance to the ape which later solved the problem. It was only when a flash of insight combined into a new meaningful whole the hints provided by the presence of the food, the stick, the ape's arm, and by its unsuccessful attempts at reaching, that success came. Even in this very low order of invention, it was necessary to use the wisdom of past experience to escape from the bondage of past experience, and it is that which makes invention so difficult.

If the stick is placed in such a position that it and the food cannot both be seen at one time, many apes cannot solve the problem. With very simple minds, all elements of the solution must be visually present at one time for the clues to be sufficiently strong for the new whole to be conceived. Even with skilled engineers and scientists, the influence of such a favorable setting is great.

Pasteur Discovers Preventive Inoculation.

Often the accidental proximity of the component elements of the solution have been the prelude to great invention. Thus just before he discovered preventive inoculation, Pasteur's illness caused him to neglect a culture of chicken-cholera microbes that he was studying. In this study, he had already observed that those few chickens which became infected and yet somehow survived the disease were thereafter immune against further infection. When after illness he returned to work he found that the undernourished microbes had lost most of their vitality. Thus by chance, the knowledge that infection gave immunity against reinfection and that protracted malnutrition produced attentuation of microbial vitality were brought to his mind when it was actively engaged in seeking means of preventing infection.

This enabled him to see a startling possibility. Might it not be possible to find means of so attentuating the microbes which caused disease that a person could be inoculated with them with no risk and only minor discomfort? Then might not the fact that the person had had the disease even in a mild form render him immune to it thereafter? He inoculated his chickens with the enfeebled microbes. The chickens displayed only the mildest symptoms of cholera. But thereafter they were immune to

* "The Mentality of Apes," W. Kohler, Harcourt, Brace, New York, N. Y., 1925, pp. 32-40.

infection by germs of full virulence. Preventive inoculation had been discovered.

Most experimenters would have thrown the neglected microbes out as unsuitable for experimental use. Even if they had bothered to examine them under the microscope the feebleness of the microbes would have made their unfitness for experimental use seem all the more clear. Only to a mind peopled with vivid concepts of disease, and sensitive to implications quite outside its own expectations, would the apparently disconnected elements of this problem come together as a richly meaningful whole.

Unquestionably, this favorable setting occurred by chance but as Pasteur so often said, "Chance favors the mind which is prepared," and well might he have added, "but only if the mind has retained its freedom and resourcefulness in spite of the discipline of preparation." It is the need of this combination of discipline and freedom that makes scientific and engineering invention so difficult.

Invention of the Automatic Loom by Northrop.

Let me take a concrete example. About 1860, loom manufacturers began trying to devise an automatic loom. By 1890, all the elements of an automatic loom had been devised except that no way had been found automatically to thread the bobbin yarn through the eye of the shuttle. Without this, an effective automatic loom could not be made.

One day a machinist named Northrop, who had been working on spinning machines, went to his chief and said he thought he could invent an automatic loom. At some time in his work a flight of inventive intuition taking off from his background of knowledge of spinning had given him a vision that the problem of threading the shuttle could be solved if the old conception of threading the yarn through the circular shuttle was abandoned and instead the yarn was wound into an open eyelet.

This inventive insight was followed by a long period of experimentation until he devised a shuttle in which a curved entrance into the eyelet was so patterned as to correspond with the whirl of the yarn as it came off the spindle. Then by merely anchoring the yarn and sliding the shuttle across the loom, the yarn as it unwound from the spindle wound itself into the eyelet. Thereafter, the faster the shuttle flew back and forth in the loom, the faster the yarn unwound and the more the twist of unwinding held the yarn in the shuttle eyelet.

With Northrop, as with the ornithologist who recognized the flycatcher, clues were meaningful which to others had been without meaning in spite of years of searching. Everyone who had ever worked on looms knew that the shuttle yarn whirled into a cone as it came off the spindle, but did not think of it in relation to the problem of threading the shuttle eye. But with Northrop, the nature of his past work caused him to approach this problem from an angle other than that from which his predecessors had looked. This enabled him to see the old factors in a new light.

As this case illustrates, the process of logical engineering development often comes to an impasse because the engineer is following principles that worked at the beginning but have come to the end of their utility. He is like a man who has planned to climb a mountain and has followed a trail which leads him near to the top but not necessarily into the best position to attain the top. To go on, he must see the mountain as a whole, recognize that he must go back and start from another point of departure in order to surmount the obstacle. Thus, inventive insight often involves seeing that the problem itself is not what has been assumed, but is of another sort involving another angle of approach.

To do this is difficult especially for the highly trained mind. For just as a man who has made the wrong turn on a detour finds it hard to make himself return to the place of deviation, so it is hard for anybody who has long studied and worked along one line of approach to try another. This is particularly true if the needed idea involves a fundamental change. In that case, to make the other approach involves more than going back and thinking from another angle. It involves reorganizing the complex structure of knowledge which he has developed with such care and pains.

The situation is very like that of the electrical current in downtown New York. In that part of the city, due to the fact that it was one of the places first wired for electric lights, direct current has been provided from the start. Every electrical device is adjusted to it. If Mayor La Guardia were to state the unquestioned fact that for ordinary commercial uses alternating current is better than direct current, and add that "effective next week alternating current will be installed in downtown New York," a storm of protest would break out. Most electrical devices would be useless until adjusted to this fundamental change.

Just as with direct-current installations, the greater and more intricate the structure of thought involved, the greater the difficulty of change. With a man who for a long time has followed one angle of approach, a fundamentally new approach may even involve changing from an expert who is working in an area where he is highly skilled and completely at home into a novice who is feeling his way. Thus the approach to the problem of drying ink through internal chemical reaction instead of mechanical drying meant that all the work on that problem which the mechanical engineer had done and the ability which it had given him were made obsolete. The problem was no longer even in his department. The internal resistance to this is inevitably great. Every habit of thought and every desire involved unite in opposition.

If others are similarly placed, the united opposition may be insurmountable even after the discovery is a well-established fact. Thus, when Pasteur, looking through his microscope at what had been known for decades —the appearance of active microbes in the course of disease—saw this not as the result of disease but as its cause, he rendered obsolete much of existing medical science. Since he had no formal medical education, he had little internal resistance to overcome. But the announcement of his discovery marshaled against him the organized and bitter opposition of the whole medical profession. Their thinking processes and their practices had all been developed on a different stem. To accept the new point of view involved giving up most of their previous ways of thought and practice, admitting that they had been wrong, and then starting the slow painful path of learning new ones.

The influence of established scientific opinion upon the inventor through the power of suggestion is strong even when overt opposition does not deter him. To invent he must so free his mind from the influence and the logic of his own past and that of his fellows, that his unconscious intuition can see clues which he and others have overlooked, and build out of them novel conceptions.

Often in order to get sufficiently away from the effects of stereotypes of thought he must utilize some external means of releasing his mind from too self-conscious and logical control. Thus new insight often comes to an inventor when he is relaxed or diverted; when he is out walking, when he is working on something else, when he is about to go to sleep, or when he wakes up in the night. Only then is his mind sufficiently relaxed to free insight from bondage to the habitual.

Faith an Essential Driving Force.

When, as is often the case, the initial insight is vague and obscure—just a hint itself that something lies in a new angle of approach—the inventor must have great determination and patience in order to keep on until he has achieved success. What brings this about? Certainly not inborn "sticktoitiveness" alone. When, for instance, Pasteur's insight told him that the germs might play a major part in disease, he kept on largely because of faith—because he was convinced that, if his intuition were true, the power of medical science to cure disease would be tremendously changed and the welfare of mankind immeasurably served.

His determination, however, rested upon more than this—it rested upon his conviction that if *he* persisted, *he* could probably accomplish the result. He knew his experimental method. He was skilled in it and aware of his skill. He saw the relation of his particular abilities to the solution of his problem. It was because he not only had faith in the value of what he might discover, but faith in himself as the person to make the discovery that he kept on. Little inventive persistence occurs unless both aspects of faith are present; and as with Pasteur, faith is one's insight and in oneself is essentially a by-product of technical understanding and high intuitive skill. The pathway to developing the drive that keeps one going until success is the slow hard one of developing the wisdom and skill that make success possible.

Quite apart from the driving force of faith is that of inner tension. To make this clear let me again cite a very humble example. When a person has seen an acquaintance whom he once knew well but cannot place, he feels a sense of strain. He has a clear image of the man he has seen. He knows that he has somewhere in his mind a clear configuration of who he is. Both are there. But he cannot bring them together into a common conception. This makes him restless. He keeps turning the problem over in his mind even when he is trying to think of something else. The more nearly he seems to "get it," the more strain he feels. Then in some moment of relaxation just who the person is comes to him in a flash and he has a great sense of relief.

So with incipient invention. If the knowledge of the inventor and the clues which will bring the invention into being have been brought nearly into position to provide the inventive insight, the inner tension will be strong. The man will keep turning the problem over in his mind. He will not be able to get away from it by day or night. As he nears his goal he will become increasingly excited. He will be impatient of the interruption of his work even for sleep or food. It is no wonder that the sudden release of such inner tension is often described by inventors as a "flash."

Intuition Versus Logic.

Curiously, in spite of the invaluable part that intuition plays in human affairs, it is common for people, especially if technically trained, to consider it as of a lower order than logical thought and to pride themselves on being exclusively logical thinkers. Logic is so much more tangible and systematic, and so easy to put into words, and hence to teach, that mute unconscious intuition is decried as unintellectual and emotional.

What are the facts? Shakespeare, Bach, Pasteur, Edison, in their moments of greatest originality were not logical but intuitive. The genius of a great musician in sensing the structure of a composition, the insight of a great doctor in diagnosing an obscure disease, and the insight of a great inventor are intellectual attainments of as high order as any logical deduction.

True, insight can also be of a very low grade. Even apes can exercise it. But as Vannevar Bush, in developing his integrating machine, has shown, complicated logical processes can be performed even by a machine.

Nor is intuition more emotional. Unquestionably, as when a thirsty man in a desert suddenly sees a mirage, emotion and desire can corrupt intuition. But emotions also lead reason astray. Indeed the process of logical reason is so effectively subject to emotional manipulation that self-deception is known the world over as "rationalization."

But isn't scientific logic quantitatively exact while intuition is quantitatively indeterminate? It is. But if one examines the application of quantitative science to reality, one finds that the exactness of scientific logic is limited. Logical quantitative methods are accurate only in regard to abstractions or when applied to "abstract materials;" to unnatural creations of man such as iron and steel; beams and railroad rails; things from which all the variables of nature have been carefully excluded.

When one deals with normal natural phenomena the realistic precision of intuition is often greater than any which abstract quantitative methods can provide. This is of course true in identifying the workmanship of an old master or recognizing a long-lost friend. But even with so simple a phenomenon as a batted ball, no scientist or mathematician can predict its flight as it leaves the bat with either the speed or the precision of a major league fielder who sees the ball hit, turns back, and runs almost to the exact place where it alights, integrating by intuition the whole complex of unmeasurable variables. In much of modern life, the most effective understanding is when scientific quantitative accuracy and scientific intuition both play a part.

It is important to bear in mind the high intellectual quality, the unemotional soundness, and the realistic integrity and precision of refined insight. For insight is the father of invention far more indisputably than necessity is its mother.

Psychological Requirements of Inventiveness.

What then, to recapitulate, are the psychological requirements of inventiveness and the factors which favor them?

First, it is essential that the inventor "know his stuff." If his inventions are to be of use in modern industry he must be well equipped with scientific understanding. This doesn't mean that he should have endless rules and formulas at the tip of his tongue. Such inert knowledge will be a burden to him. What he needs to have is a vital knowledge of basic scientific principles and have developed realistic creative skill in using them, by practice and experience.

As a second qualification the inventor must have, and must continuously develop against the encroachment of custom and habit, the capacity to construct bold effective and varied configurations and concepts out of the clues which lie concealed in technological problems. The more he knows, the more difficult it will be for him to do this. What I fear most as an obstacle to inventiveness in industry is that as a young engineer pursues the long slow path of education and experience, he succumb to the past as New York has to its direct-current installations.

The most important way out of this dilemma is to make education and experience more fundamental. If in school and at work the resourceful young engineer is required to shun memorized formulas and to go back to basic principles for the solution of his problems, the process of developing technical competence will in itself give him the basic understanding and way of thought essential for inventiveness. When logic thus goes

back to fundamental origins, and is in that sense original, logical development and inventive insight come close together. Both make possible fundamental shifts to more profound bases of approach. Even so, with truly baffling problems, logical exploration can only do the groundwork for the flight of intuitive insight. If such insight is to occur, the process of education in school and industry must make provision for its exercise. Again and again the formal paths of logic must be abandoned and the relaxation provided which permits insight unconsciously to find its way.

Change of Mental Scene.

Another aid to the freeing of inventiveness from the confining influence of learning is to assign to resourceful young men creative problems outside of their customary field of work. Such an alien approach, as it so often has with great inventions, may provide a suggestive setting which will lead to a solution of the particular problem. But more important is the fact that it will tend to free able minds from the confining adhesions of their customary thought.

Since inventiveness is not confined to major discoveries such change may be more frequently provided in less formal ways. Any engineer or executive who is worth his salt should show inventive talent in his daily work and be required to do so. Even in reading, rumination, and talk, the power of creative intuitive thought should be exercised and developed. Much can be done to stimulate this by bringing young engineers into frequent serious-minded association with able men of other backgrounds. By so doing they will be caused to see their own problems from new slants and in the light of different practices and principles.

Far more than senior executives usually realize, what they do, what they are, and what they admire in their subordinates determines what those young men who work for them do, think, and become. Through example, they can keep their juniors in the habit of resourceful yet sound intuitive judgment. They can inspire in them admiration for and determination to imitate the people who have the daring and ability to abandon traditional ways of thought, and who while doing so arrive at conclusions which logic may at first refute but later accept. A third psychological factor favoring inventiveness is thus wide-spread admiration for sound intuitive thought.

A fourth is determination. But if a young man has vital and profound scientific understanding, if he is resourceful in its intuitive use, and if he admires and cultivates these qualities, I have little fear that he will lack the faith which gives determination. Few men who have attained high scientific intuitive skill through the hard path of study and experience lack the determination to use their skill well. By example and by the work they assign and by the qualities to which they give recognition, executives can do much to cause their outstanding subordinates to revere and study fundamental science and also to cherish and develop the intuitive spark within them which enables them to put this science into richly creative use. Young men of promise can do even more to accomplish this for themselves.

HIGH ALTITUDE ROCKETS FOR METEOROLOGICAL RESEARCH.—This article by Willy Ley and Herbert Schaefer presents their views that there are strong arguments for the use of instrument-carrying rockets in meteorological research. Mr. Ley was Vice-President of the

German Rocket Society (Verein fur Raumschiffahrt) from 1928 to 1933. Mr. Schaefer was one of the designing engineers on the Society's proving ground. Both went into voluntary exile soon after Hitler became chancellor of Germany.Their article is reprinted from The Military Engineer, November, 1943.

The recognition of the fact that the atmosphere of the earth forms only a thin shell of gases around the planet is of comparatively recent date. It seems that the astronomer Johannes Kepler was the first to state explicitly that the celestial bodies are separated by an airless space, in his posthumous book *Somnium*, or, by implication, that the atmosphere does not extend beyond a certain distance from the surface.

Before Kepler it had been taken for granted that air could be found at any altitude, that the atmosphere, to use modern phraseology, extends at least through the whole solar system. Only under that assumption was it possible that Icarus could literally come too close to the sun. Nor was there any thought of diminishing density, thus Lucian could speak of a strong wind which carried a vessel to the moon.

It was only under these assumptions that it was logical to believe that the moon was really as smooth as a polished shield and that the spots were dirt that had condensed from "vapors" in the air. Even much later one can find an occasional reference to birds with singed wings, taken not as evidence of forest fires but as evidence that that particular bird had shared the fate of Icarus.

The astronomer Kepler, who discovered and established the laws of planetary motion, could not use such a resistant medium as air in interplanetary space and was consequently forced to conclude that an atmosphere was a surface phenomenon, astronomically speaking. How he reconciled this conclusion with the established philosophical belief in the *horror vacui* is not quite clear, but that idea was to be dispelled soon anyway.

Exploration of the Upper Atmosphere.

Kepler died in 1630 and one year later Francesco de Lana-Terzi was born, the theoretical inventor of the balloon. The invention can be found in the sixth book of the second volume of his great work *Magisterium Naturae et Artis*, published in 1686. The description is not at all vague, but quite clear and definite: "It is certain that one can construct a vessel of glass or other material which could weigh less than the air contained therein; if, then, one exhausted all the air this vessel would be lighter than the air itself and, therefore, it would float on the air and ascend."

Lana was right in theory, but greatly underestimated the practical difficulties of this suggestion. Still, his theoretical discussion had pointed out what should be done and the idea took hold. Suggestions were made to use "the thin air from mountain tops" instead of a vacuum, and when Henry Cavendish announced the low specific weight of his "inflammable air" in 1766 (it was, of course, hydrogen) the invention of the balloon was virtually made. It took hardly another eight years until the first hydrogen balloon was built by Professor Charles. In the meantime the Montgolfiers had succeeded in demonstrating their first hot air balloons.

The very first hydrogen balloon that took to the air, on December 1, 1783, carried a barometer and a thermometer; the first balloon ascent specifically devoted to meteorological research took place one year later, almost to the day, November 30, 1784. Dr. John Jeffries of Boston, Massachusetts, then residing in London, had fathered that expedition in more than one way.

Before that, some atmospheric exploration had been initiated by Pascal and by Perier, Pascal's brother-in-law. The latter can be said to have started the science of meteorology by carrying a so-called Torricellean tube to the top of the Puy-de-Dome (Auvergne, France) on September 19, 1648. The height of the Puy-de-Dome is only 1,460 meters (4,790 feet), but it was a beginning. And just as Perier's experiment was but the first of a large number of similar "expeditions," the first ascent of Dr. John Jeffries was followed by many others. Often enough the observers risked their life in the open gondolas of those days, but the balloon permitted the exploration of conditions in the atmosphere away from mountain peaks. In addition, it permitted the exploration of altitudes far above the peaks, at least, of the more accessible mountains.

But it took almost a century from Dr. Jeffries' first flight until somebody thought of doing this kind of research work by means of automatically registering instruments, carried by small, unmanned and inexpensive balloons. The first suggestion of this kind seems to have been made in 1879 by Brissonet. Brissonet's suggestion had only one drawback: self-registering small meteorological instruments did not exist. The German, Assmann, made a similar suggestion soon after Brissonet; it was an independent suggestion and it had the same drawback as Brissonet's.

In 1891 Georges Besancon and Gustave Hermite began to build such instruments, and Assmann did the same. And on March 21, 1893, the first *ballon sonde* or sounding balloon was released in France. It reached an altitude of 15,000 meters or about 49,200 feet. A German sounding balloon, released on April 27, 1894, reached 21,800 meters, or 71,500 feet, approximately the same altitude as that attained many years later by the manned stratosphere balloons: the American *Explorer II* and the Russian *Stratostat*. But, meanwhile, the sounding balloons had broken that early record again and again; at present the highest recognized altitude of a sounding balloon belongs to one that was released near Hamburg on September 8, 1930. It attained 35,900 meters or 117,750 feet.

There is no doubt that most of our knowledge of the physical state of the upper atmosphere has been derived from the data gathered by the instruments of these balloons. There is also no doubt that this knowledge is very important in many ways . . . but unfortunately there can also be no doubt that the method by which these data have been obtained is not very efficient.

A stratosphere balloon is enormously expensive and can not be counted upon to be good for more than one ascent. The sounding balloon never lasts for more than one ascent (it keeps ascending until it bursts, the instruments being lowered to the ground by means of a parachute) and is, fortunately, not very expensive, but it suffers from other drawbacks. The ratio of losses (of the instruments) is high, since they are subject to drift both during the ascent by balloon and during the descent by parachute. It is often all but useless to release balloons on a day of zero visibility which results in interruptions in the series of data gathered.

More important, however, than the high ratio of losses and the all too frequent days when the balloons stay "grounded" is the fact that they rarely even approach the normal ceiling of aircraft equipped with superchargers. Those record figures of 70,000 and 100,000 feet are very decidedly record figures; not even 30,000 feet can be counted upon as definitely attainable.

It is obvious, therefore, that another type of instrument carrier, about as inexpensive as a sounding balloon, but more reliable and with greater climbing ability, would be preferable by far.

Such a carrier exists: the rocket.

Rockets as Instrument Carriers.

The main advantage of rockets as compared with other possible means, such as stratosphere balloons, high altitude airplanes and, of course, sounding balloons, is the absence of the concept of "ceiling" or "peak altitude." A peak altitude exists only as far as a rocket of specific design and size is concerned, since it is possible, at least theoretically, to design and construct a rocket for any desired altitude.

The present goal is, of course, more modest; it may be described as a preliminary design for a rocket which will normally and regularly attain the record altitudes of the best sounding balloons. In other words, rockets with a vertical range of a minimum of 10 and a maximum of 25 miles above sea level.

It is probable, as will become apparent later on, that such rockets could be designed, tested, and finally produced without an undue amount of preliminary research, especially since the first experimental design could be based upon the rather large amount that has been done along these lines in foreign and, at present, enemy countries before Hitler's rise to power. It is furthermore probable that the price of such rockets would be low while the expenditure required for one ascent would compare favorably with the price of a sounding balloon.

Fortunately the performance of a rocket—what may be called the "natural" performance—is favorable to the intended use as an instrument carrier. The instruments will not be subjected to shocks of any kind and special designs will, therefore, not be necessary.

A rocket ascends by means of the reaction against a stream of gases ejected through a downward pointing exhaust nozzle. Those gases are created by the rapid combustion of suitable substances, the exhaust speed depending mainly on the nature of the fuels used. The impulse can be calculated by the formula

$$P \times t = m \times c \text{ or } P \times dt = dm \times c$$

which, when solved for P, results in the formula

$$P = c \times \frac{m}{t} \text{ or } P = c \times \frac{dm}{dt}$$

In these formulae P stands for the recoil, t for the time, and m for the ejected mass, while c has been chosen as symbol for the exhaust speed.

It is obvious that the recoil varies with a variation of the ejected mass as well as with a variation of the exhaust speed. At the same time it is easy to see that an increase of the exhaust speed c is more important than an increase of the amount of mass ejected per second.

When P becomes larger than the weight of the rocket, the latter begins to rise with a steady acceleration. (Or rather the acceleration would be steady if the rocket maintained its original weight.) But the steady consumption of fuel causes a steady decrease in weight which results in an increased acceleration if the factors c and dm/dt remain the same, in other words, if the thrust does not fall off. It may be added here that this is the condition that usually prevails, the thrust remains the same until the supply of fuel is exhausted. Both acceleration and speed of ascent are, of course, modified by the action of air resistance which leads to very interesting relationships as will be shown later.

At any event, the rocket will rise—generally speaking with slowly increasing acceleration—until the fuel supply is exhausted. This is known as the ascent under power at the end of which the rocket has attained the highest speed of the whole ascent. Because it then has a considerable kinetic energy the ascent under power is followed by the "free ascent" which continues with steadily decreasing speed until the kinetic energy

of the rocket has been used up by the combined actions of air resistance and gravity. If the ascent of a rocket be compared with the ascent of an antiaircraft shell the ascent under power is the equivalent of the part of the shell's trajectory that is inside the gun barrel while the "free ascent" compares with the real trajectory from the muzzle to the peak of the altitude reached by the shell. The descent that follows afterwards is a normal free fall, unless broken by means of a parachute.

The instruments of our hypothetical rocket would not have to stand a higher strain than that caused by the acceleration of the rocket, an acceleration which is not very high, well within the limits of strains caused by ordinary use.

Preliminary Experimental Work.

Systematic rocket research was carried out for the first time more than a century ago by the British general Sir William Congreve. It led to the introduction of the British bombardment and incendiary rockets which bore Congreve's name. After the successful and very spectacular use of these rockets against Boulogne, Copenhagen, and Danzig the "new" weapon (actually incendiary rockets had existed as far back as 1250 A.D.) was imitated by virtually every other country in Europe and also by the United States. Since Congreve succeeded in keeping the secret of the manufacture of his rockets (there is reason to suspect that he created a few additional "secrets" where none existed) each Ordnance Department had to conduct its own research. But while the period of the war rockets of the nineteenth century is a highly interesting chapter in military history, the research then conducted can not very well be termed scientific; it was pure craftsmanship, the improvements were created by the trial and error method.

The first man to conduct scientific rocket research in the modern sense of the word was Dr. Robert H. Goddard who began his career as a rocket researcher by improving the United States Army signal rockets during World War I. Doctor Goddard worked with various powders and powder mixtures as rocket fuels for some time; this part of his work has been described in his book *A Method of Reaching Extreme Altitudes*.[1] Later Doctor Goddard began to work with liquid fuels for rockets; a kind of preliminary account about this phase of his work was published under the title of *Liquid-Propellant Rocket Development*[2] in March 1936.

While this work was in progress nothing authoritative was published about it, so that the experiments of European experimenters, carried out over the same interval of time, roughly from 1929 to 1933, were not at all influenced by Dr. Goddard's results and methods. Of these many European experiments, which were carried out by an astonishingly large number of individuals, firms, and organizations, only two series of rocket tests are important since all the others either gave up too quickly after preliminary failures or else lacked the funds or facilities to carry on until fairly conclusive results were obtained. These two series of importance are the experiments of Dr. Eugen Sanger of Vienna, made mostly at Vienna University, and the experiments of the German *Verein fur Raumschiffahrt* (German Rocket Society), carried out on the society's own proving ground near Berlin. Dr. Sanger's experiments were made mostly during the years of 1930 and 1931 while the experiments of the *VfR* were begun in 1930 and continued without interruption until Hitler became chancellor of Germany.

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[1] Smithsonian Misc. Coll. Vol. 71, No. 2, 1919.

[2] Smithsonian Misc. Coll. Vol. 95, No. 3.

Dr. Sanger worked with light fuel oils and gaseous oxygen under very high pressures, up to 300 atmospheres, and after a few preliminary experiments arrived at a kind of standard design for his "rocket motor." It was made of steel and spherical in shape with a very long exhaust nozzle attached at one side. A second steel shell surrounded the spherical "combustion chamber" and that part of the exhaust nozzle closest to the combustion chamber, the so-called throat. The fuel oil was forced into the space between the outer shell and the combustion chamber and entered the latter through a fine injection nozzle. The oxygen line went from the tank directly into the combustion chamber, passing through the outer shell. The fuel thus acted as fuel as well as a cooling agent. Such rocket motors produced steady runs of more than 20 or even 25 minutes with a recoil of about 60 pounds.

Dr. Sanger only performed test stand runs; he never progressed to the design of an actual rocket.

The first phase of the work of the *VfR* bore a natural resemblance to that of Dr. Sanger. There also were some preliminary designs of different shapes and sizes and there also emerged a best type which then became more or less standard. The *VfR* used liquid oxygen only, since compressed gaseous oxygen could not very well be carried in a rocket because of the great weight of a suitable container. The rocket motors were made of spun sheet aluminum and the parts welded together. The shape of the combustion chamber can best be described in saying that a sphere was cut in halves and a cylinder two radii high was inserted between the two hemispheres. The exhaust nozzle was comparatively short and the cooling jacket surrounded the whole motor, being welded to it around the muzzle of the exhaust nozzle. The diameter of the combustion chamber was approximately 55 millimeters and the whole motor did not weigh more than 120 grams, about 4½ ounces (before the valves were attached).

This type was designated as "160/32" which meant that it consumed 160 grams of gasoline and liquid oxygen per second, delivering a thrust of 32 kilograms (about 70 pounds). Actually most motors were "150/30;" the designation had been taken from one of the first models which happened to be somewhat more powerful than the later average.

Most of the early difficulties encountered were caused by the lack of experience in handling liquid oxygen; as soon as several members of the experimenting crew had acquired some practice it was easy to run off up to half a dozen tests per day. The fuel used was ordinary low-octane gasoline; since rockets prefer "knocking" fuels benzene might have been better. It was tried only once with the anticipated result: it got stuck in valves and pipelines because of its high freezing point and the close vicinity of the ultra-cold liquid oxygen. Alcohol, theoretically almost as good as gasoline for rockets, was repeatedly tried later and performed satisfactorily. It was mainly a kind of inertia which caused the majority of all tests to be conducted with gasoline as a fuel.

After sufficient experience with rocket motors on the test stand the first rockets were built. They consisted essentially of two tanks, one for the fuel and the other for the oxygen, a rocket motor which had been tested on the test stand at least twice (later three times), and a container for the parachute. The tanks consisted of seamless magnesium alloy tubing, known under the trade name of *Elektron*, welded at both ends. The inside diameter was 35 or 45 millimeters, depending on the size of the rocket, and their length was sufficient to hold enough fuel for a 45-second run. For reasons that will be explained presently the tanks could not be charged to full capacity.

Thin aluminum tubes ran to the bottom of the tanks at one end and to the injection nozzles of the combustion chamber at the other end. Some-

where between tank and rocket motor (the precise place again differing with the model) the feed lines were blocked by valves which had to be closed against powerful springs; as soon as the pins holding them were removed the valves snapped open.

When a rocket was to be tested the gasoline tank was filled first, to less than a third of its full length. The remaining space was filled with compressed nitrogen gas. The space between cooling jacket and combustion chamber was filled with water and the filling hole closed by a loosely fitting plug which often had a center hole. The last job, after inserting a pyrotechnic firing capsule into the nozzle was to fill the oxygen tank to about two-thirds of its full length. It was sealed with a tightly fitting plug and the oxygen quickly built up pressure inside the tank which could be read off a small pressure gauge. When the pressure inside the oxygen tank equalled the nitrogen pressure in the gasoline tank—30 atmospheres—the firing capsule was ignited electrically. Undisturbed it would have burned for 10 seconds, but immediately afterwards both valves were released and snapped open. The stream of oxygen and gasoline droplets caught fire immediately which blew the firing capsule out of the nozzle. The rocket began to rise about a quarter of a second later.

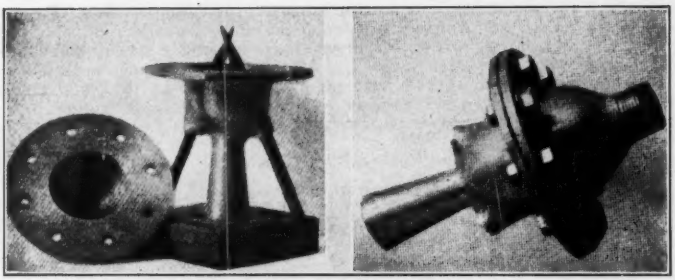
At the peak of the flight a small parachute was forced out of its container by a timing mechanism which had been set according to calculation and had been started automatically by the passage of the rocket through the launching rack which consisted of a simple structure of aluminum angles and was about 20 feet high.

The first rockets of this type had a pulling rocket motor and both tanks side by side, with the parachute container attached to the lower ends of the tanks. But because of the unequal weight of the tanks these rockets tended to assume a horizontal position after a few seconds of fairly straight ascent. For this reason the two tanks were later placed in line with the parachute container at the bottom and the rocket motor set in a four-pronged fork (two of the prongs were the fuel and oxygen lines) at the top. These rockets performed well, reaching an altitude of close to 1 mile. All in all some 400 test runs were made during the three and a half years of experimental work and about 120 ascents. Many rockets were used five and six and even ten times.

There were also some very heavy models, the largest weighing 125 kilograms fully fueled, but there is no need to discuss these models here since they can not be regarded as a prototype of a meteorological rocket.

But a few words of explanation for the bottom position of the parachute may be in order. When the rocket is in free ascent it is under no acceleration whatsoever, nor is it during the descent, when it follows the gravitational attraction of the earth freely. Consequently, there is hardly a way to operate the parachute release at the precise moment when peak altitude is reached. Pendulums do not work, nor do pendulum-like devices such as small quantities of mercury in a glass tube. Strictly speaking, there is a minor difference between free ascent and free descent, caused by the action of air resistance, but this was not considered sufficient to be reliable. For this reason, a clockwork mechanism was used to release the parachute, set to work at the calculated time interval between firing and standstill before falling back. But there was the possibility of error, of course, or of minor malfunctioning. If the parachute had been placed in the nose of the rocket, on top of the rocket motor, very precise functioning would have been required.

If the mechanism should release the parachute three seconds too late while the rocket was falling back a nose parachute would hardly be effective, for on account of the tail fins, it was to be expected that the rocket would turn over, falling motor first and then, in case of such a belated



MODEL OF ROCKET MOTOR FOR 65-POUNDS THRUST.

Pencils indicate the flow of the two liquids, gasoline and liquid oxygen.

For actual use flange would be cut off and the two halves welded together.

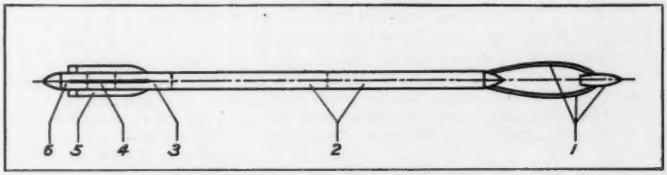
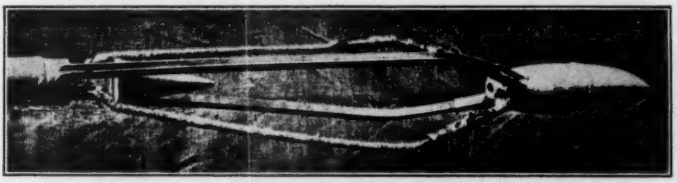


DIAGRAM OF METEOROLOGICAL ROCKET.

1. Rocket motor and fuel lines; 2. Oxygen and fuel tanks; 3. Space for steering device if needed; 4. Space for instruments; 5. Fins; 6. Space for parachute and ejecting mechanism.



HEAD OF THE ROCKET, SHOWING FUEL LINES AND STRUTS.

release, the rocket would fall into its own parachute, presumably wrapping itself up in the parachute. The same would happen if the timing mechanism should release a nose parachute while the rocket was still climbing. None of these dangers existed with a tail parachute. If it were released too early it would just have cut down the total altitude; if it were released too late nothing at all could happen. Consequently, a tail parachute was used exclusively and the releasing mechanism set for a slightly belated release.

A Meteorological Rocket.

These experiments permit a preliminary design of a meteorological rocket which would consist, beginning at the top, of a rocket motor similar to the type described, a tubular oxygen tank, a tubular fuel tank, an additional tubular container to house the meteorological instruments, the parachute with releasing mechanism, and, if required, a steering mechanism.

So far only the last of Doctor Goddard's rockets had a steering mechanism, presumably consisting in the main of a small gyroscope and acting on movable vanes to prevent deviations from a vertical ascent. Films taken of these experiments show that the steering mechanism functioned well. The rockets of the *VfR* did not have steering mechanisms because they proved to climb almost vertically just because of the arrangement of the component parts. (Doctor Goddard's last rockets had a pusher motor, situated at the bottom of the rocket.) If meteorological rockets with a nose drive, as the pulling motor is usually called, will require a steering mechanism or not, can hardly be decided without experiments.

If they do not need a steering mechanism it will be a saving in weight as well as in price. If they do, it would not be difficult to develop such a mechanism. It can be very simple because it would have to work for only a few minutes and would not need to be more accurate than to prevent deviation from the vertical line of more than 5 degrees. Similar mechanisms for airplanes are much more accurate and work for many hours, but are much too heavy.

The instruments would have to be such that they fit into a tube about 4 inches in diameter, of any required length, and must not weigh more than one kilogram or 2.2 pounds. Judging from the instruments carried by sounding balloons the available space as well as the weight limit are more than ample. The whole arrangement would be such that the rocket, when returning, is suspended from the instrument cage by a thin rope some 30 feet long, while the instrument case is suspended from the parachute. The rocket would then touch the ground first and relieve the parachute of its weight for an additional drop of 30 feet which would cause the instrument cage to touch the ground with a very low falling speed. Because of this it would be possible to operate with a fairly high falling speed for the whole assembly, which would help to cut down drift. The ascending rocket will not be subject noticeably to drift, unless the ascent takes place in a high wind.

Computation of Probable Altitudes.

In order to compute the altitudes that can be attained by such rockets it was necessary to assume a specific model and to calculate its performance step by step.

For purposes of comparison two such rockets were assumed, both of dimensions much smaller than rockets actually built in the past. These rockets were called Type I and Type II. Type I was assumed to weigh

9 kilograms empty, which includes the parachute, et cetera, to hold 10 kilograms of fuel and oxygen, and to carry instruments weighing 1 kilogram. The fully fueled rocket would then weigh 20 kilograms, its diameter would be 100 millimeters, about 4 inches.

Rocket Type II was assumed to have a diameter of 120 millimeters, to weigh 20 kilograms empty, hold 20 kilograms of fuel and oxygen, and carry 6 kilograms of instruments.

The ratio between oxygen and fuel would depend on the nature of the fuel. In the case of gasoline the equation for complete combustion reads as follows:

1 kg. C_8H_{18} + 3.5 kg. O_2 = 4.5 kg. CO_2 and H_2O and in the case of alcohol:

1 kg. C_2H_6O + 2.08 kg. O_2 = 3.08 kg. CO_2 and H_2O The exhaust gases are harmless in either case, consisting of carbon-dioxide and water vapor. The theoretical exhaust speeds for these two cases are 4450 meters per second for gasoline and oxygen and 4180 meters per second for alcohol and oxygen. During the *VfR* experiments an exhaust speed of 2000 meters per second was attained regularly and sometimes 2200 meters per second. Dr. Sanger reported exhaust speeds over the range from 2500 meters per second to over 3000 meters per second.[3] The exhaust speed of both rockets, Type I and Type II, was assumed to be 2000 meters per second in some calculations and 2500 meters per second in others.

It was assumed that the fuel consumption would be constant and that the thrust P would, consequently, be constant too. The initial accelerations were assumed to be 1 g, 2 g and 4 g, respectively. Even the highest thrust required in these assumptions has been surpassed experimentally.

The ascent was calculated in the following manner:

Air resistance according to Siacci [4]

$$D = 338 \times R^2 \times i \times \gamma \times f(v)$$

Air density according to the Hohmann formula.[5]

The factor $i = 1.2$[6]

Ascent under power:

$$\text{Acceleration } a = \frac{(P-W-D)}{W} \times g.$$

$$\text{Speed } v_2 = v_1 + \frac{(P-W-D)\ g}{W} \times \Delta t$$

$$\text{Altitude } s_2 = s_1 + \frac{v_1 + v_2}{2} \times \Delta t$$

Free Ascent:

$$v_2 = \sqrt{v_1^2 - \frac{2g}{W_1}(h_2 - h_1) \times (D + W_1)}$$

Explanation of symbols:

D in kilograms	Air resistance.
R in meters	Radius of rocket.

[3] In *Raketen-Flugtechnik* (Munich 1933) and in personal correspondence.

[4] Cranz and Becker: *Ballistics.*

[5] *Die Erreichbarkeit der Himmelskorper*, Munich 1925. The formula reads: $\gamma/\gamma_0 = (1 - h/400{,}000)^{4.9}$.

[6] for shells $i = .865$.

$i = 1.2$	
γ in kg/m^3	Density of the air.
$f(v)$	According to Siacci (Cranz-Becker: *Ballistics*).
a in m/sec^2	Acceleration.
a_s in m/sec^2	Effective acceleration at beginning of ascent.
v in m/sec	Speed of rocket at any given moment.
s in meters	Altitude attained under power.
h in meters	Altitude attained "free."
$H = s + h$	Total altitude.
P in kilograms	Thrust.
W in kilograms	Weight of rocket at any given moment.
W_0 in kilograms	"Take-off weight" of rocket.
W_1 in kilograms	Weight empty, but including instruments.
t in seconds	Time.
g	Assumed to be the same at any altitude attained by rocket with the value of 9.81 m/sec^2.

The computation of the ascent under power was made with intervals of $\Delta t = 2$ seconds. For this interval W, γ and $f(v)$ were assumed to be constant with the value which is correct for the first instant of this interval. The error thus created is very small and has the tendency to cancel out during the computation. The free ascent was calculated with intervals (of altitude) of $(h_2 - h_1) = 500$ meters. The variables were treated in the same manner.

The results of these rather tedious computations are condensed in the following two tables:

TABLE I.

$W_0 = 20$ kg; $W_1 = 10$ kg; $W_F = 10$ kg; $R = .05$ m.

The symbols are the same as used in the formulae, with the following additions:
W_F Weight of fuel and liquid oxygen.
G_s Fuel (and oxygen) consumption per second.
v_{max} Highest velocity attained by the rocket during the ascent.
T Time required for complete consumption of fuel and oxygen.

#	c (m/sec)	a_s (m/sec^2)	P (kg)	G_s (kg/sec)	T (sec)	v_{max} (m/sec)	s (km)	h (km)	$H = s+h$ (km)
1.	2000	1 g	40	.196	51.0	460	12.9	7.2	20.1
2.	2000	2 g	60	.294	34.0	515	10.2	7.1	17.3
3.	2000	4 g	100	.491	20.4	590	7.2	7.0	14.2
4.	2500	1 g	40	.157	63.7	575	18.7	12.2	30.9
5.	2500	2 g	60	.236	42.5	625	14.7	11.3	26.0
6.	2500	4 g	100	.392	25.5	695	10.3	10.3	20.6

Table Ia—Case No. 5 under the assumption that the rocket is fired from the top of a high mountain instead of sea level.

5a*	2500	2 g	60	.236	42.5	790	20.8	22.6	43.4
5b†	2500	2 g	60	.236	42.5	830	22.3	26.4	48.7

* Height of mountain assumed to be 4 km in case No 5a.
† Height of mountain assumed to be 5 km in case No. 5b.

TABLE II.

$W_0 = 46$ kg; $W_1 = 26$ kg; $W_F = 20$ kg; $R = .06$ m.

#	c	a_s	P	G_s	T	v_{max}	s	b	H
1.	2000	1 g	92	.451	44.4	465	10.8	7.8	18.6
2.	2000	2 g	138	.676	29.6	545	8.8	9.1	17.9
3.	2000	4 g	230	1.130	17.7	640	6.2	9.5	15.7
4.	2500	1 g	92	.361	55.4	560	15.8	12.3	28.1
5.	2500	2 g	138	.541	57.0	650	12.8	14.5	27.3
6.	2500	4 g	230	.902	22.2	745	9.0	14.0	23.0

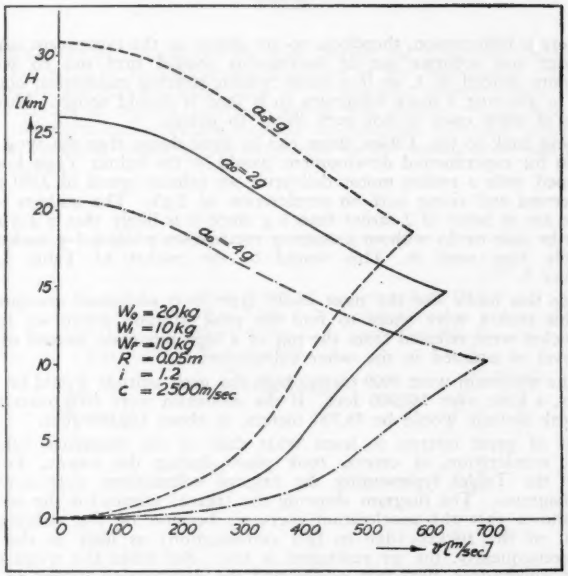
It can easily be seen that the exhaust speed c and the time of burning are the most important factors influencing the performance. Rockets of equal weight with an exhaust speed and a burning time one quarter larger and longer than that of other rockets attain altitudes some 40 per cent higher. This holds true for rockets of about the size of the assumed types only.

If the rockets were to ascend in a vacuum it would hold generally true that the rocket ascending with the highest acceleration would reach the highest altitude.

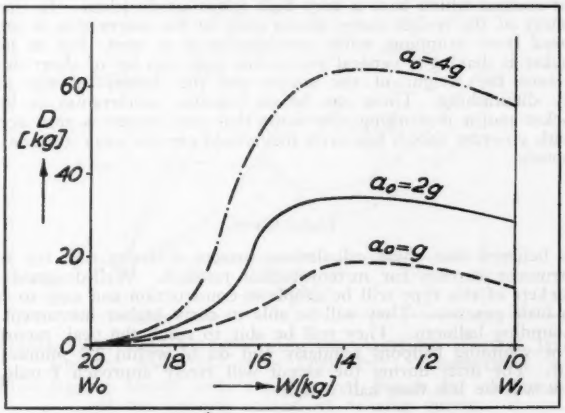
Precisely the reverse is true for meteorological rockets. The Tables show that the lowest altitudes in each case belong to the rockets which start out with an acceleration of 4 g's. The highest altitudes in each case belong to the rockets starting out with the lowest acceleration. The reason for this performance is the action of air resistance. High acceleration rockets reach a high speed when still comparatively low above the ground and they, therefore, encounter a high air resistance in comparatively dense layers. Low acceleration rockets do not attain high speeds until they have reached comparatively high altitudes where the air is less dense.

But even the slowest (and highest) climb does not take longer than a maximum of two minutes, as compared to at least half an hour for sounding balloons.

Another interesting point is the behavior of the heavier rockets of Type II. When compared with the lighter rockets of Type I the heavier rockets do not quite attain the same peak altitude when the acceleration is only 1 g at the beginning of the powered ascent. But those that start out with accelerations of 2 g's and 4 g's, respectively, get higher than the lighter rockets accelerating at the same rate. They get higher, that is, if we look only at the last column of figures in the tables, showing the total altitude. They do not get as high as the others during the powered ascent. But, being heavier and possessing a higher kinetic energy, they show a greater power of penetration and thereby make up for their lower-powered altitude by a longer free ascent. This shows that weight-saving is not quite as important as may be thought at first glance. If the weight of Type I, for constructional reasons, should turn out to be 25 kilograms instead of 20 kilograms the loss in peak altitude would be less than might be expected.



VELOCITY CURVES FOR THREE ROCKETS WITH DIFFERENT ACCELERATIONS. Showing total altitude H and velocity at any point during the whole ascent, both "powered" and "free." Peak velocity is reached at the moment the fuels give out. Although the rocket with the highest acceleration attains the highest velocity the one with the lowest acceleration attains the greatest height because its performance, as regards air resistance, is more advantageous.



AIR RESISTANCE (d) CURVES FOR THE THREE ROCKETS.

There is little reason, therefore, to cut down on the instrument load if a compact and accurate set of instruments should turn out to weigh 2 kilograms instead of 1, or if a more reliable steering mechanism could be built in allotting 3 more kilograms to it than it should weigh. However, either of these cases is not very likely to occur.

Going back to the Tables, there can be little doubt that the type to be chosen for experimental development would be the lighter Type I rocket, equipped with a rocket motor delivering an exhaust speed of 2500 meters per second and rising with an acceleration of 2 *g*'s. The authors of this article are in favor of 2 rather than 1 *g* since it is likely that a 2-*g* rocket might be able to do without a steering mechanism while a 1-*g* rocket presumably may need it. This would be the rocket of Table I, case Number 5.

Since this looks like the most likely type, two additional computations for this rocket were made to find the peak altitude above sea level if the rocket were released from the top of a high mountain instead of from sea level as assumed in the other calculations.

If the mountain were 4000 meters high the peak altitude would be 43,400 meters, a little over 140,000 feet. If the mountain were 5000 meters high the peak altitude would be 48,700 meters, or about 160,000 feet.

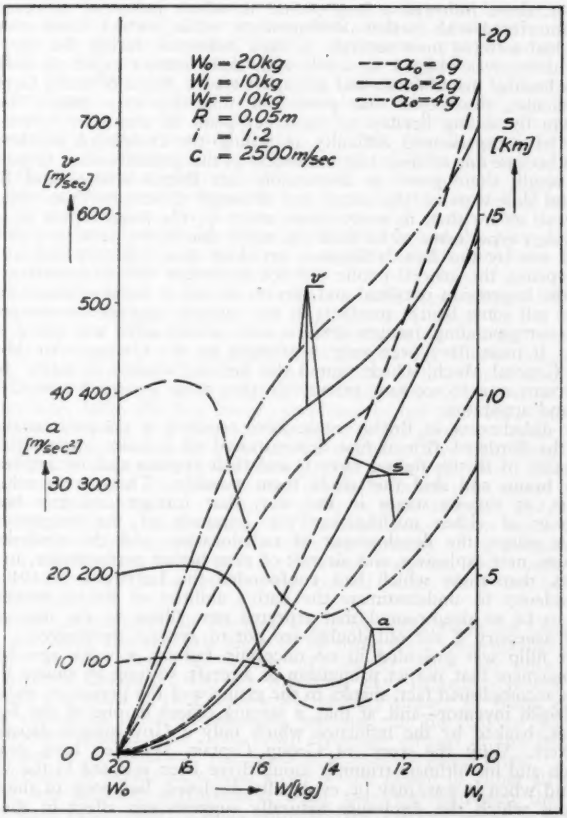
It is of great interest to learn what shift of the numerical values of speed, acceleration, et cetera, took place during the ascent. For this reason the Tables representing the original calculations were converted into diagrams. The diagram showing the (three) curves for the acceleration shows that the acceleration increases because of the reduction of weight of the rocket (due to fuel consumption) as long as the speed and, consequently, the air resistance, is low. But when the speed of the rocket approaches 1000 feet per second the acceleration begins to drop. This drop is most pronounced for the rocket with an initial acceleration of 4 *g*'s and proportionally less pronounced for the two others. The 4-*g* rocket is only 1100 meters above the ground when it approaches that critical speed, while the 1-*g* rocket does not approach it until it has climbed to 4500 meters.

Theoretically it is possible that, in the case of a poorly designed rocket, the acceleration will drop to zero; as a matter of fact it does happen to powder rockets which have a very high initial acceleration. In that case the power of the rocket motor shows only to the extent that it prevents the *speed* from dropping, while acceleration is at zero. But as long as the rocket is making a vertical ascent this state can be of short duration only, since the weight of the rocket and the density of the air are steadily diminishing. There can be no negative acceleration as long as the rocket motor is working; the worst that can happen is zero acceleration with constant speed, but even this would cut the peak altitude down enormously.

Conclusion.

It is believed that these calculations present a strong case for rockets as instrument carriers for meteorological research. Well-designed liquid fuel rockets of this type will be simple in construction and easy to handle after a little practice. They will be able to carry higher instrument loads than sounding balloons. They will be able to reach the peak record altitudes of sounding balloons regularly and do so within 2-3 minutes after take-off. The drift during the ascent will rarely approach 1 mile, normally it will be less than half a mile.

Nothing can be done about the drift during the parachute descent, but there are ways of facilitating the discovery of the instruments. It would be easy to incorporate some kind of pyrotechnic device which releases smoke clouds at regular intervals during the descent. Also there exist small and light radio devices, running on flashlight batteries, which produce a humming sound in the earphones of the search crew and permit triangulation.



ACCELERATION (a), VELOCITY (v), AND ALTITUDE (s) FOR THREE ROCKETS ASCENDING WITH DIFFERENT ACCELERATIONS.

This diagram covers only the powered ascent, for continuation of the velocity curves see velocity curves.

JET PROPULSION.—This editorial from Engineering (London), January 14, 1944, gives some of the background of recent press releases regarding jet-propelled aircraft.

During the last war, many and notable technical and scientific advances were made in British design offices, factories and laboratories, the disclosure of which induced comforting reflections that this country still retained that skill in making and doing which had been responsible to so great an extent for the national prosperity in the latter part of the Nineteenth Century, and the early years of the Twentieth. After the war, however, there followed a long period in which industrial depression at home interfered with further developments; while, abroad, those countries which had suffered most severely in their industries during the war years found themselves obliged to work with their utmost vigor to build up anew a healthy manufacture and an export trade. Some of them, Germany in particular, discovered that good and relatively easy money was to be made by selling licenses to construct plant in countries where they would have experienced difficulty in selling the completed articles; and Britain became one of their best customers in this profitable line of business. As a result, there grew an impression that British science and British technical skill were on the wane; and although those who knew the facts were well aware that, in many cases, much of the good repute in which the foreign types came to be held was really due to the development work carried out by the British licensees on their own initiative and at their own expense, the general public did not appreciate this all-important fact. Thus the impression persisted and spread, so that it became almost impossible to sell some British products in any quantity against the competition of the corresponding foreign articles, even where price was not a ruling factor. It must have been very heartening to the German Air Mission, under General Milch, which visited the British "shadow factories" before the present war, to see how extensively they were equipped with German tools and apparatus.

The disbelievers in British capabilities received a salutory awakening when the Battle of Britain first demonstrated on a really impressive scale the quality of British fighter aircraft and their engines and, by implication, of the brains and skill that made them possible. There were other reminders, at various stages in the war, that foreign countries had no monopoly of either qualification—the conquest of the magnetic and acoustic mines, the development of radiolocation, and the evolution of new guns, new explosives, and aircraft of even higher performance, in some respects, than those which had confounded the Luftwaffe in 1940; but the tendency to underestimate the native abilities of British technicians seems to be so deep-rooted that repeated new fillips to the imagination appear necessary if the old doubts are not to reassert themselves.

That fillip was provided in no uncertain fashion, a week ago, by the announcement that the jet propulsion of aircraft, so long an elusive dream, was an accomplished fact, thanks to the genius and the persistent endeavors of a British inventor—and, at that, a serving officer in one of the fighting Services, backed by the influence which only a Government department can exert. Why the story of Group Captain Whittle's long years of research and his ultimate triumph should have been released in the way it was, and when it was, may be eventually disclosed; but none of the many questions which the disclosure naturally suggests can affect in the least the fundamental fact that he has initiated a development that may be of decisive importance in the war and may take rank, in the opinions of future judges, with the pioneer work of any of those of his fore-runners who devised and developed the reciprocating internal-combustion engine, and perhaps even with that of Parsons on the steam turbine.

The announcement was made simultaneously in London and Washington, on the evening of January 6; by the Air Ministry and the Ministry of Aircraft Production in this country, and by the War Department in the United States. It stated simply that "jet-propelled fighter aircraft have successfully passed experimental tests and will soon be in production," and proceeded to outline the course of the development of this new aircraft engine and the machine in which it was tried. Group Captain Whittle's interest in the subject of aircraft propulsion was aroused in 1930, when he was taking an instructor's course at the Central Flying School, but it was not until 1933 that he began work on the first engine. Four years were spent in this preliminary development, but he achieved his aim and the first engine ran successful trials in April ,1937. The Air Ministry was aware, of course, that the experiments were in progress, and decided, in 1939, to construct an aircraft to be propelled by the Whittle engine. The order was placed with the Gloster Aircraft Company, Limited, the engines being built in a special factory by Power Jets, Limited, to whom Group Captain Whittle was lent by the Royal Air Force. The first successful flight was made in May, 1941, the pilot being the late Flight Lieutenant P. G. Sayers, chief test pilot of the Gloster Aircraft Company. The official statement observed with reason, at this point in the narrative, that "the greatest credit should be given to Group Captain Whittle for this fine performance, for it was his genius and energy that made this possible"; to which might be added an appreciation of the part played by Flight Lieutenant Sayers in taking up a machine which must have differed widely from any that he had ever flown.

Two months later, to continue with the official statement, full information about the new engine was disclosed to General H. H. Arnold, commanding the United States Army Air Force; who, realizing its possibilities, asked at once for a specimen to be sent to the United States. The engine which had made the first flight was sent to the works of the General Electric Company in September, 1941, and there a number of similar engines were built, the first being completed in less than six months. At the same time, the Bell Aircraft Company were given an order for an aircraft to be powered by two of the engines. The first flight in the United States was made in less than a year from that date. Since then, it is stated, several hundred successful flights have been made in this country and in America, "many of them at high altitudes and extreme speed, all without a single mishap." Plans have now been made for the production, in both countries, of a sufficient quantity of jet-propelled aircraft to enable them to be used for training purposes, and a number of these machines will be allotted to the United States Navy for additional experiments.

The first American-built machine made its initial flight on October 1, 1942, this being also the first flight to be made by a jet-propelled combat aircraft. It was piloted by Robert M. Stanley, chief test pilot of the Bell Aircraft Company. On the following day, Brigadier-General (then Colonel) Lawrence C. D. Craigie made military history by being the first Army officer to fly a jet-propelled machine. In the United Kingdom, jet-propelled aircraft have been flown by many officers of the Royal Air Force and by a number of test pilots of aircraft-building firms.

Naturally, no details have been given of the design of either aircraft or engine, nor of their performance, other than the statement that many flights have been made "at high altitudes and extreme speed." Since the propulsive reaction is created within the machine, and does not depend upon the ability of propeller blades to "grip" the air, the new type of engine should be particularly suitable for high-altitude flying; but, as the rate of discharge of the gas constituting the jet depends upon the rate at

which the fuel is burned, and so upon the rate of air intake, a compressor is an essential feature. This much may be deduced, in fact, from the remarks contributed by Group Captain (then Squadron Leader) Whittle to the discussion on Dr. Adolf Meyer's paper on "The Combustion Gas Turbine: Its History, Development, and Prospects," read before the Institution of Mechanical Engineers on February 24, 1939.

Group Captain Whittle remarked that neither Dr. Meyer nor the previous speakers in the discussion had said much about the gas turbine in relation to aircraft, which he regarded as the most hopeful field for it, because of the increase in its overall efficiency and in the power per pound of air per second, in rising from the ground to the stratosphere; for example, assuming a turbine efficiency of 85 per cent and a compressor efficiency of 70 per cent, a maximum temperature of 1100 degrees C. absolute, and an adiabatic temperature rise in the compressor of 200 degrees, C., the overall efficiency at ground level would be 18.45 per cent and the brake horsepower per pound of air per second would be 59.3; but, at 35,000 feet, the efficiency would be 27 per cent and the brake horsepower, 98.5. Of course, the total power output fell off, due to the reduced air density; but so did the power required for a given flying speed. The overall efficiency of a reciprocating internal-combustion engine, however, fell off to such an extent that an unsupercharged engine could hardly overcome its own mechanical losses at 40,000 feet. The big flying boat had reached a point, Group Captain Whittle continued, where it was limited in development by its power plant; at present (1939), no suitable engine would produce more than 2000 Hp. in one unit, although power units with an output of 5000 Hp. to 10,000 Hp. were needed. Moreover, there were, as was well known, strong arguments in favor of flying at very great heights. The combination of these two factors resulted in a demand in the aeronautical world for efficient gas turbines.

It is known that experiments with jet propulsion, along these general lines, were in progress in Germany, Italy and France some years ago. The Italian machine, the Caproni-Campini, flew, though its performance compared badly with that of machines with more orthodox power plants. The advantages offered by successful jet-propulsion are so great, however, that it is a safe assumption that interest on the Continent has not relaxed, even though the construction of actual aircraft may be in abeyance. These advantages include not only the higher efficiency mentioned above, but also improved aerodynamic efficiency of the aircraft; smaller and lighter landing gear (the lofty under-carriages of aircraft are only necessary to prevent the large-diameter propellers from striking the ground); and a power plant that should be relatively simple to manufacture. There are disadvantages also, no doubt, but it is equally certain that they will be reduced in time. Even as it is, however, the jet-propulsion motor appears to have possibilities for other applications than in aircraft, though this seems to be the most promising. Engineers and scientists alike will watch its further development probably with even more interest than they have accorded to aircraft in past years.

BOOK REVIEWS.

THE PRACTICAL DESIGN OF WELDED STEEL STRUCTURES, BY H. MALCOLM PRIEST. PUBLISHED BY THE AMERICAN WELDING SOCIETY, 33 WEST 39TH ST., NEW YORK, 18, N. Y.

This small volume brings up to date a paper published by the author in 1933. It is a reprint of a paper published in September, 1943, with additions from another article by the same author.

Beyond mentioning the author and publisher, little need be said about the book. It is short, terse, understandable, readable, and covers the field thoroughly. Where details of codes or specifications are required to complete the presentation, a specific reference is made in the text. So, also, where space does not permit in the scope set for himself by the author and he feels that the reader might wish or need to pursue the item further, a suggestion as to additional reading is inserted.

In all a very valuable book.

THE RADIO AMATEUR'S HANDBOOK. PUBLISHED BY THE AMERICAN RADIO RELAY LEAGUE, INC., WEST HARTFORD, CONNECTICUT. $1.00.

The 1944 edition of this well known publication is now available. This is the 21st Edition. It is devised to meet the conditions of 1944, a period of total war and is a distinct contribution to the war effort.

The volume has gotten larger than its predecessors because of the need for including more and more information to permit the Handbook to fulfill its mission. To meet wartime needs,

the editor has in his own words put some emphasis on the "How-it-Works" part of the book than on the "How to Make it."

There are 169 pages devoted to "Principles and Design," 213 pages on "Construction and Data" and 8 on "Operating and Traffic."

LANGE'S HANDBOOK OF CHEMISTRY. Compiled and Edited by Norbert Adolph Lange, PhD. Published by Handbook Publishers, Inc., Sandusky, Ohio.

This is the fifth edition of Lange's well known tool for the scientific worker. The work has been revised and enlarged. Included are the following tables which were not a part of the previous edition:

Periodic Table (Deming)
Flammable Liquids
Flame Temperatures
Plastics
Fluorescence of Chemicals, Minerals and Gases
Water for Industrial Use

In addition, thirteen tables have been extended or completely rewritten. The most important of these is the complete revision of the table of Physical Constants of Organic Compounds which has been increased to include 6507 compounds with a corresponding list of more than 3500 synonyms.

The task of revising and reprinting this manual in wartime merits the thanks and welcome reception of the scientific world.

ASSOCIATION NOTES.

ERRATA: The article on the *Lafayette* Salvage published in the November, 1943, issue of the JOURNAL stated, on page 648, that the vessel had been launched at LeHavre. This is incorrect, as she was launched at St. Nazaire.

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The following have joined the Society since the publication of the November, 1943, JOURNAL:

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